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NAVAL SHIPS TECHNICAL MANUAL

CHAPTER 9470

PUMPS

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NAVAL SHIP SYSTEMS COMMAND, WASHINGTON, D. C. 20360

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CHAPTER 9470 — PUMPS

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SECTION I. RECIPROCATING PUMPS

Part 1. General

9470.1 BASIC PRINCIPLES

1. The pumping action in any reciprocating pump is dependent upon the positive displacement or the fluid pumped by a piston or plunger. The capacity of the pump is, therefore, determined by the area of the piston and its rate of travel. In order to obtain a practical machine, some method of reversing the direction of the pistons is required. In the direct-acting steam pump, this is accomplished by the steam valves and valve gear; in power pumps, this is accomplished by use of crank and connecting rods.

2. The ability of the pump to produce pressure is dependent upon the ratio of total steam force (steam pressure per unit area x area of steam piston) to total liquid force (pump head x area of liquid piston). In order that pumping may occur, it is necessary that the steam force exceed the liquid force by an amount which slightly exceeds the various mechanical and hydraulic losses encountered. The basic principles for steam pump operation are shown in figure 9470-1.

9470.2 TYPES OF PUMPS

1. Direct-acting reciprocating pumps are classed as follows:

a. Horizontal or vertical.

b. Single or duplex. A single pump has one liquid piston or its equivalent single or double-acting plunger; a duplex pump has two liquid pistons or their equivalent single or double-acting plungers.

c. Single or double-acting. A single-acting unit pumps on one direction of piston travel only whereas double-acting units pump on both strokes. Direct-acting steam pumps are usually double acting.

2. Direct-acting steam pumps are conventionally described by stating the steam cylinder diameter, the liquid cylinder diameter, the length of stroke, horizontal or vertical (H or V), single or duplex (S or D), and single or double-acting (SA or DA). Thus a pump identified as 11 x 8 x 18 VSDA has an 11-inch steam cylinder, 8-inch water cylinder, 18-inch stroke, and is a vertical single double-acting pump.

9470.3 USAGE IN NAVAL SERVICE

1. Direct-acting steam pumps in naval service are installed on combatant ships for the following services:

a. Emergency feed.
b. Fire and bilge.
c. Fuel oil tank stripping and bilge.

2. On auxiliary ships a great number of applications are still served by steam reciprocating pumps, including:

a. Auxiliary feed.
b. Standby fuel oil service.
c. Fuel oil transfer.
d. Auxiliary circulating and condensate.
e. Fire and bilge.
f. Ballast.
g. High pressure evaporator.
h. Lubricating oil transfer.
i. Cargo stripping.
j. General service.

3. Direct-acting steam reciprocating pumps are not obsolete. If the steam conditions are not too severe in pressure, temperature, or superheat, they have many features of simplicity, reliability, and economy of operation and maintenance that still warrant serious consideration for many services.

9470.4 STEAM VALVE GEAR

1. Steam valve gear for direct-acting reciprocating pumps can be classified by the following basic types:

a. The direct-acting flat face slide valve. This type is suitable for low steam pressures, in the order of 150 to 200 psi and below.

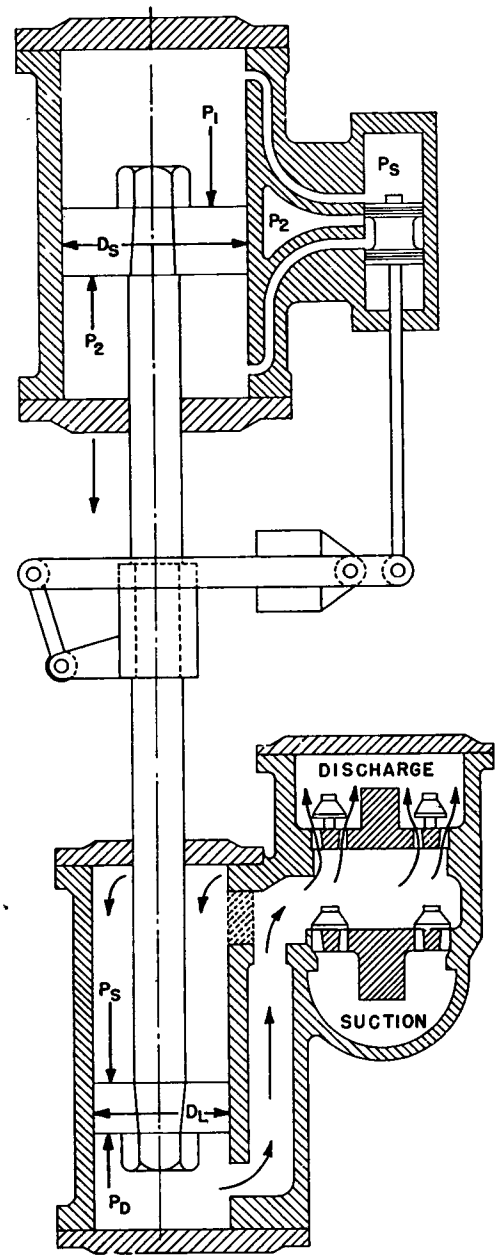
b. A steam thrown piston main valve actuated by a flat faced pilot or auxiliary valve. This type is suitable for moderate steam pressures possibly up to 500 psi and is frequently found on naval auxiliary ships converted from merchant ships.

c. A steam thrown piston main valve actuated by a piston type pilot valve. This design is capable of operation at the higher steam pressures and temperatures encountered on naval vessels. The piston main and pilot valves eliminate unbalanced loads and minimize steam leakage and wear.

2. Valve gears of various manufactures differ to the extent that a brief description of several of the designs is desirable. The descripitions following cover a majority of the pumps installed in naval service.

9470.5 DAVIDSON VALVE GEAR

1. The Davidson balanced piston-type valve gear is illustrated in figure 9470-2. Pilot valve "A" is direct-actuated



L = LENGTH OF STROKE IN INCHES.
D_S = STEAM CYLINDER DIAMETER, INCHES.
D_L = LIQUID CYLINDER DIAMETER, INCHES.
P_1 = STEAM PRESSURE P.S.I.G.
P_2 = EXHAUST OR BACK PRESSURE P.S.I.G.
P_D = PUMP DISCHARGE PRESSURE P.S.I.G.
P_S = PUMP SUCTION PRESSURE P.S.I.G. (NEGATIVE IF SUCTION LIFT EXISTS)
N = NUMBER OF DOUBLE STROKES PER MINUTE.
S = PISTON SPEED IN FEET PER MINUTE. $S = \frac{LN}{6}$
E = PUMP EFFICIENCY.

PRESSURE EQUATION

$$(P_1 - P_2) D_S^2 \times E = (P_D - P_S) D_L^2$$

CAPACITY EQUATIONS

FOR SINGLE CYLINDER, DOUBLE ACTING PUMP:

$$Q(G.P.M.) = .0064 D_L^2 LN$$
$$= .0385 D_L^2 S$$

THESE FORMULAS INCLUDE AN AVERAGE CORRECTION FOR VOLUMETRIC EFFICIENCY AND PISTON ROD DIAMETER.

Figure 9470-1. Schematic Pressure-Volume Diagram

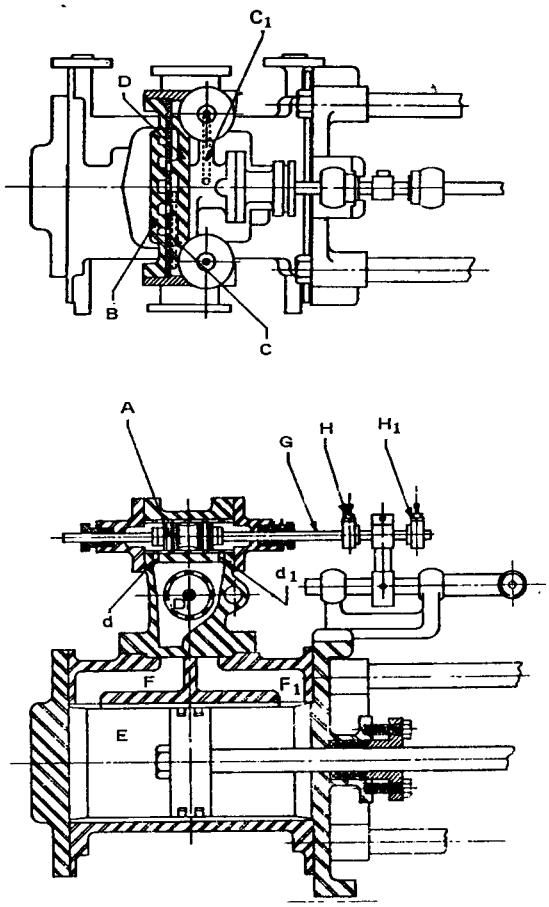


Figure 9470-2

by drive rod and valve rod "G" from the crosshead. The valve rod passes through clearance hole in center of pilot valve. Ends of pilot valve chest are under exhaust pressure only, the pilot valve having drilled balance ports end to end.

2. Steam pressure is maintained in central chamber of pilot valve "A". When valve is actuated by valve rod, inside edge uncovers the port "C" leading steam pressure to one end of main valve chest, at the same time the outside edge uncovers part "C_1" leading exhaust to opposite end of main valve chest, causing valve "D" to throw. Main valve is thrown by inlet steam, not exhaust pressure.

3. Steam is then admitted to steam cylinder "E" through port "F" and port "F_1" is opened to exhaust.

4. When piston has completed the stroke, pilot valve is actuated in opposite direction and reverse cycle takes place; port "C" is opened to exhaust, port "C_1" is opened to steam throwing main valve to opposite end of chest causing steam piston to reverse.

5. The ports "d" and "d_1" of ends of pilot chest bores are fitted with removable liners recessed on outside lead directly into main chest exhaust port and balance the pressure on pilot valve. Both main and pilot chest diameter with drilled ports around the circumference, completely balancing both piston valves. Since the main valve is horizontal, it does not have to work against gravity.

6. The length of stroke may be adjusted by means of tappets "H" and "H_1" on the threaded portion of pilot valve rod "G".

9470.6 WARREN ECLIPSE VALVE

This valve is shown in figure 9470-3 and its operation is as follows:

1. The operating lever is mounted on a fulcrum pin held by a pedestal secured to the centerpiece. At its lower end it is connected to the piston rod crosshead by a link, while near its top is mounted a pin which engages a tappet. On the valve rod link are two adjustable tappet collars. The tappet moves between these collars as the level is moved by the action of the piston rod. These tappet collars transmit the action of the tappet to the valve rod through the valve rod link and knuckle. The valve rod is connected inside the chest to the valve yoke, which sits over the stem of the main slide valve. This valve yoke throughout its entire stroke carriers with it the auxiliary valve which is set in one side. Toward the end of the movement of the valve yoke it is brought in contact with the stem of the main slide valve, slowly moving it until the auxiliary valve causes the main valve driving piston to operate. This last action takes place as follows: As soon as the valve rod begins to move, the auxiliary valve carried by the valve yoke closes the exhaust and opens the steam port of the main valve driving piston, admitting steam to the end from which it had last been exhausted. This balances the valve driving piston, which remains balanced until the main slide valve has been moved to near its central position. At this point the auxiliary valve closes the steam port and opens the exhaust port on the end toward which the valve driving piston is moving, allowing the remainder of the movement of the slide to be completed by the unbalancing of the steam pressure on the valve driving piston. This causes the pump stroke to be reversed.

2. Since it requires more energy to bring a body from a state of rest into motion than it does to accelerate that motion, it will be seen that, by mechanically moving the slide valve from a state of rest and continuing that action through approximately half of the valve's travel, when steam is exhausted from one end of the main valve driving piston, the pressure on the other end instantly and easily accelerates the movement of the slide valve. Positive action is thus obtained. The main valve driving piston is so constructed that danger of sticking is obviated.

9470.7 WARREN PISTON VALVE

The action of this valve is as follows:

1. Referring to figure 9470-4, the operating lever is mounted on a fulcrum pin held by a pedestal secured to the centerpiece. At its lower end it is connected to the piston rod crosshead by a link, and at the upper end it is connected by a pin to the valve rod link. On this link are two adjusting collars. An auxiliary lever with the lower end pivoted on a fixed pin connects to the valve rod. On this lever is mounted a tappet which is moved by the adjusting collars on the valve rod link, one located above the tappet and one below. The motion is thus transmitted from the piston rod to the valve rod, attached to which is the auxiliary piston. The action of the auxiliary piston controls the opening of steam and exhaust ports for the

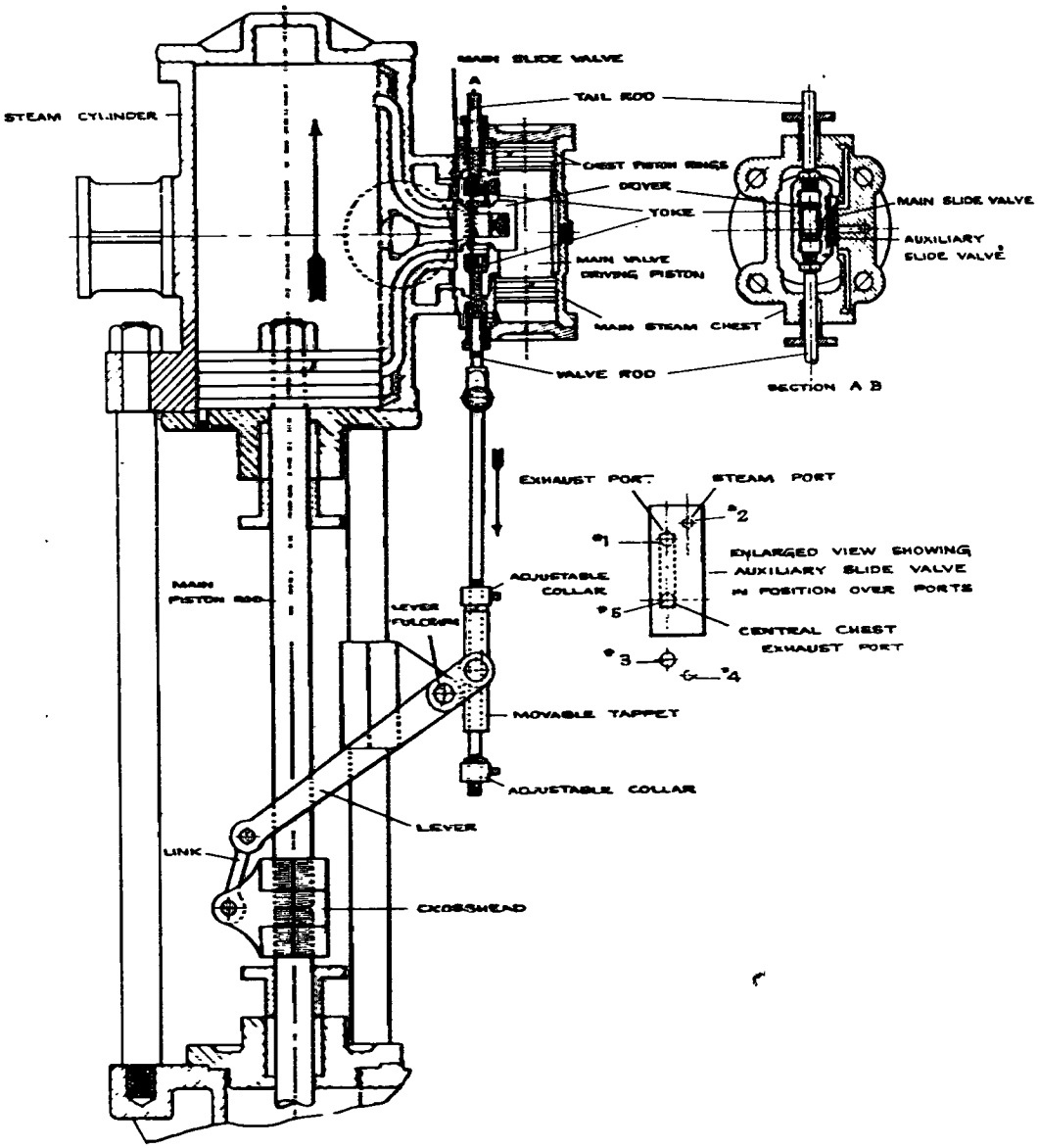


Figure 9470-3

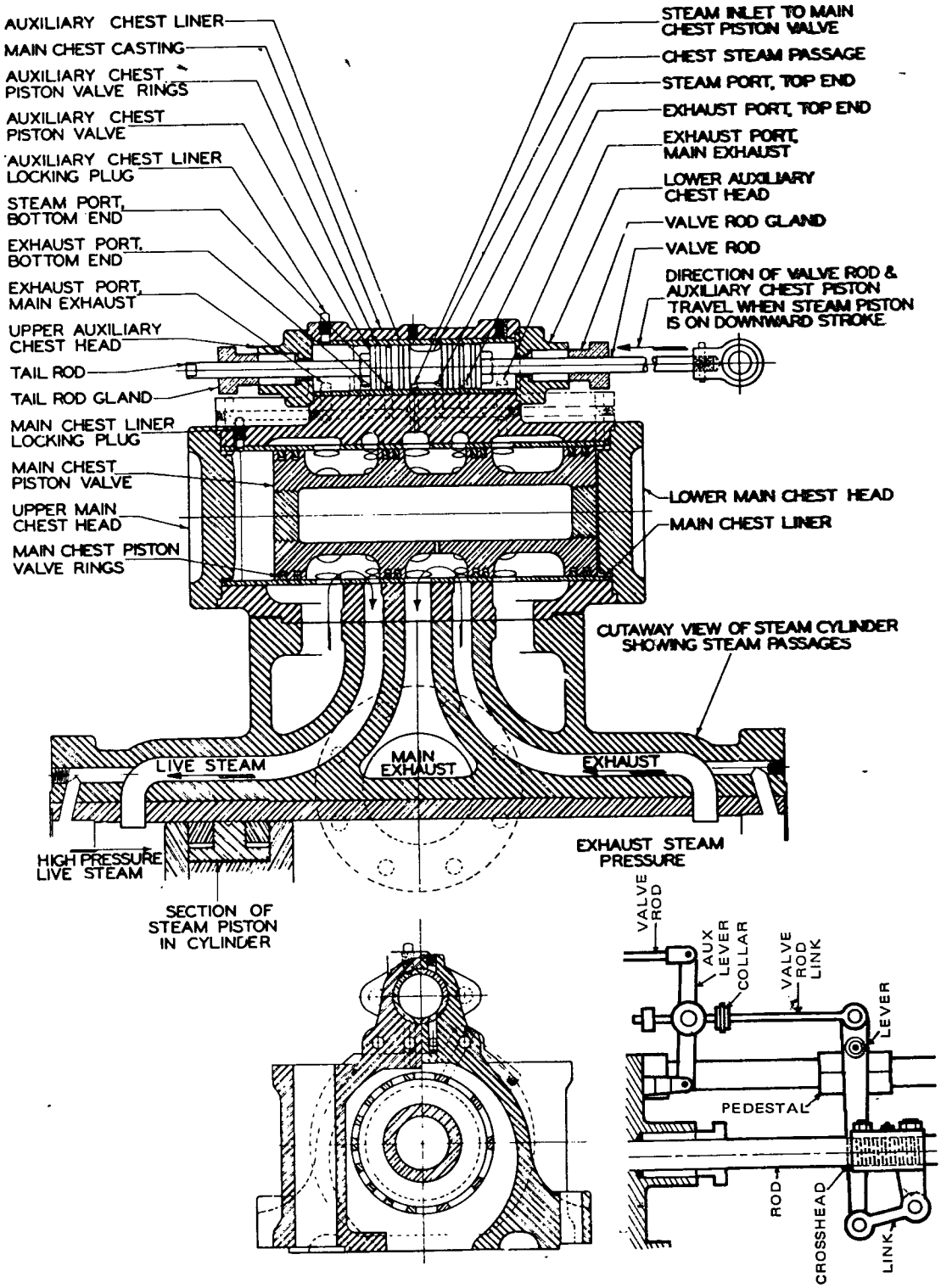


Figure 9470-4

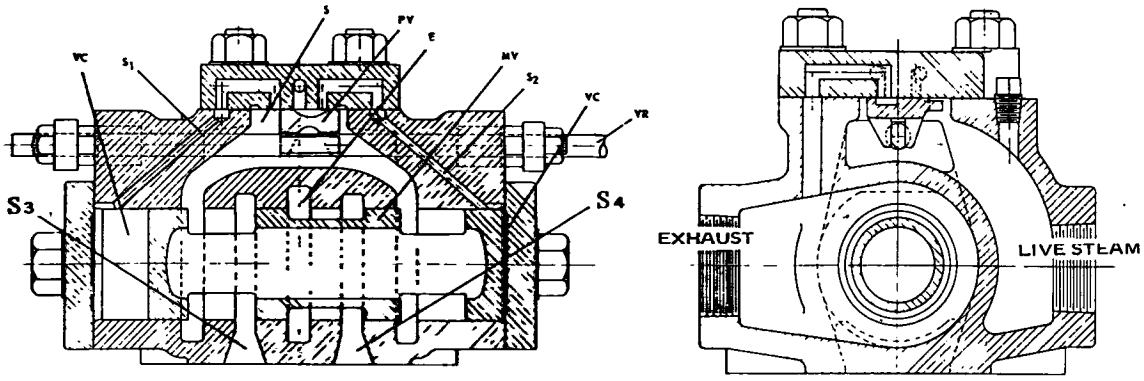


Figure 9470-5

main piston valve causing it to move and open the steam and exhaust ports to the main steam piston.

2. Referring again to figure 9470-4, the steam piston having started its downward stroke the valve rod link is moved upward and the lower adjusting collar, toward the end of the stroke, comes in contact with the tappet, thereby transmitting the upward movement to the auxiliary lever, the valve rod, and the auxiliary chest piston valve. As the auxiliary chest piston valve begins to move upward, it closes the steam port, top end, and with just a little lag closes the exhaust port, bottom end. Traveling further it opens the top exhaust port and with just a little lag opens the steam port, bottom end, thus allowing steam to reach the bottom end of the main piston valve pushing it upward to the top end of the chest, reversing the flow to the main steam piston in the cylinder.

3. The valve rod and tail rod stuffing boxes are subject to exhaust pressure only. Live steam enters only between the outside spool on each end of the auxiliary chest piston valve and the next inner spool. The exhaust is between the central spools and there is a balancing port running parallel with the auxiliary chest piston valve, connecting the exhaust passage with each end of the valve. The main chest piston valve is fitted with four snap rings. The auxiliary chest piston valve also is fitted with snap rings. Both operate in renewable liners of alloy iron which have been bored and honed. The tension of the snap rings prevents the main valve from moving when steam is turned off.

9470.8 WORTHINGTON SIMPLEX VALVE

This valve is illustrated in figures 9470-5 and 9470-6. Its operation is as follows:

1. The pilot valve PV is moved back and forth by the valve rod VR which, through links and levers, is actuated from the pump piston rod R. The pilot valve controls the admission and exhaust of steam to the ends of the valve chest VC. This moves the main piston valve MV back and forth.

2. The main piston valve controls the admission and exhaust of steam to the ends of the steam cylinder C of the pump. With the pilot valve in the position as shown in figure 9470-5, steam is admitted from the live steam space S through the left hand steam port S_1 to the end of the main piston valve forcing it to the position as shown.

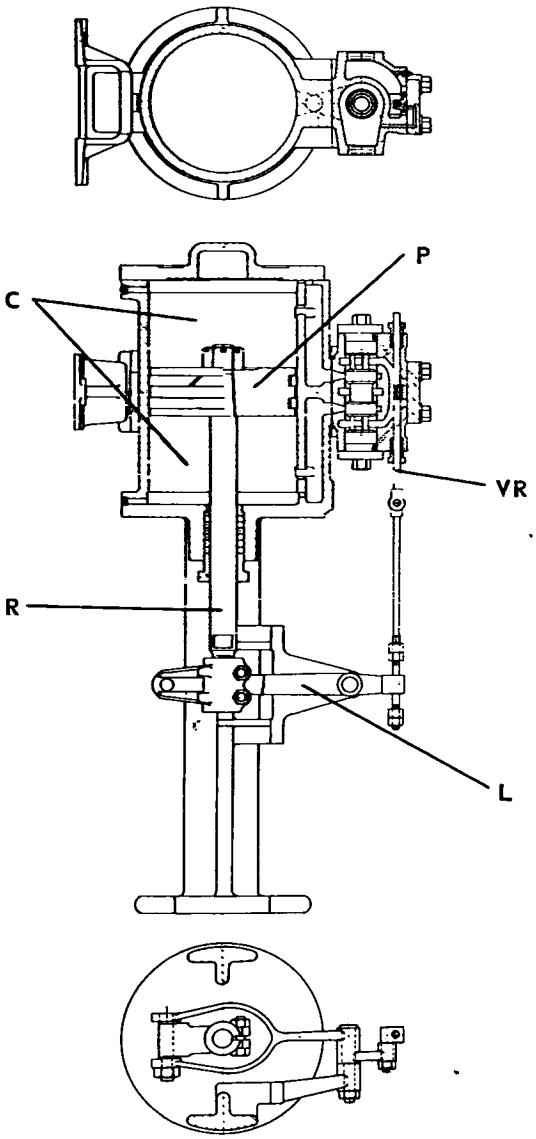


Figure 9470-6

3. With the main piston valve MV as shown, steam is admitted from the live steam space through the left hand port S_3 into the main steam cylinder C forcing the steam piston P towards the opposite end. At the same time the steam from the opposite end of the steam cylinder is exhausted through the right hand port S_4 to the steam exhaust E.

4. When the steam piston has reached the opposite end of the cylinder, the valve rod linkage L will have moved the pilot valve to the left so that live steam is admitted from the live steam space through right hand steam port S_2 to the other end of the main piston valve MV forcing it to the opposite end from that shown. In this position live steam is admitted to the main steam cylinder through the right hand port S_4 forcing the piston toward the opposite end. At the same time the steam from the opposite end of the steam cylinder is exhausted through the left hand port S_3 to the steam exhaust.

9470.9 WORTHINGTON PISTON VALVE

1. Figure 9470-7 shows the Worthington Navy-type piston valve. The reciprocation of the main steam piston P is controlled by the functioning of main piston valve V and auxiliary piston valve AV. The auxiliary valve AV is mechanically operated through lever and cranks from the piston rod. When steam is admitted through the steam nozzle, it fills the chambers marked S. There is one steam port and one exhaust port leading to each end of the auxiliary cylinder AC. The admission and exhaust of steam to and from the auxiliary cylinder is controlled by movement of the auxiliary valve AV. This causes the main steam valve V to reciprocate and admit and exhaust steam to and from the main steam cylinder MC and thus move the main steam piston P forward and back.

2. The exhaust from the main and auxiliary cylinder is also controlled by the main and auxiliary valves respectively and lead into the chamber E and out through the exhaust nozzle.

3. The strokes of the pump may be lengthened or shortened by moving the collars on the valve rod link farther apart or closer together.

9470.10 CAPACITY CHARTS

For determining the capacity of a reciprocating pump handling liquids (not air pumps), figure 9470-8 will be found convenient. The capacities indicated apply to single-cylinder pumps, in good repair, operating at full stroke.

9470.11 POWER PUMPS

Low pressure, motor-driven reciprocating pumps (power pumps) are occasionally used in naval service for fresh water sanitary, bilge, ballast, and fuel oil transfer services. For small capacities they have a better efficiency than other types and are self priming. Such pumps are generally horizontal. Both plunger and piston types are used. Their crankshafts are driven from the motor by gearing or V-belt drive. The liquid end of these pumps is similar to that of direct action steam pumps and the instructions in parts 2, 3, 4, and 5 of this section should be followed.

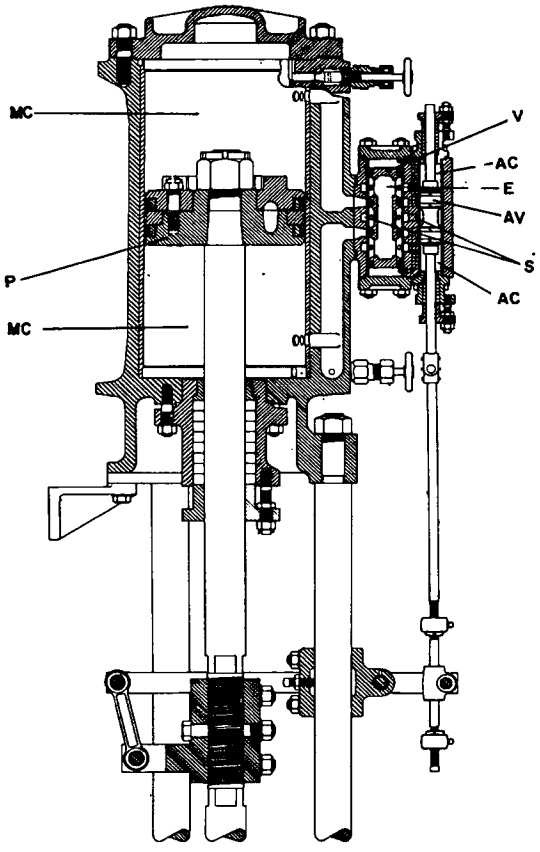


Figure 9470-7

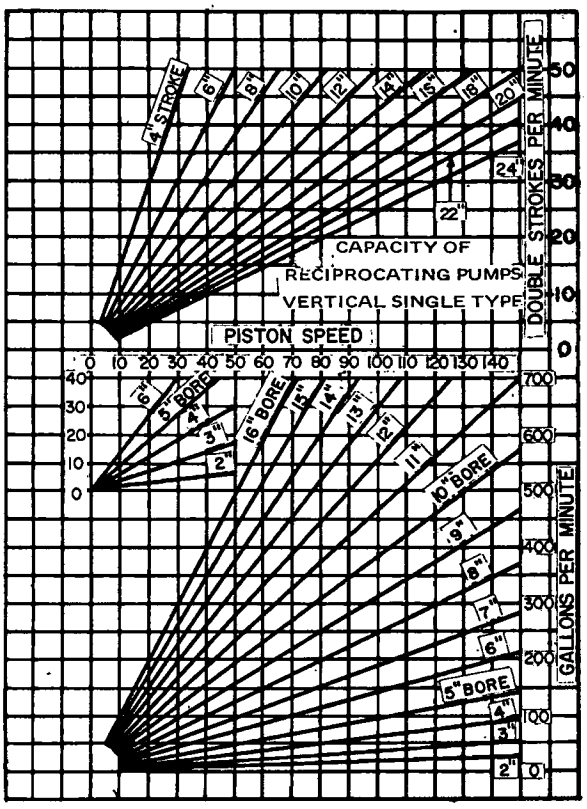


Figure 9470-8

9470.12 VARIABLE CAPACITY BOILER FEED PUMPS

Special types of plunger pumps are used for boiler feed service on a number of auxiliary ships. These pumps are variable capacity triplex units and are governed by controlling the plunger stroke by automatic devices. This permits a stepless variation in capacity from zero to maximum using a constant speed motor or turbine. The inherent high efficiency of the plunger pump for low capacity at high pressure is thus maintained over a wide range of pump output. Figure 9470-9 shows the Aldrich Groff variable capacity feed pump. Figure 9470-10 shows the general arrangement of the Worthington pump; figure 9470-11 shows the detail of the crank and eccentrics and the stroke-shifting mechanism.

Part 2. Operation

9470.21 PREPARING PUMP FOR OPERATION

The following steps should be taken before putting a pump into operation for the first time, after an overhaul, or after the ship has been drydocked:

1. Check alinement and correct if necessary. If pump is operated out of line, scoring of rods and liners will result.
2. Steam and liquid lines should be free from scale and foreign matter.
3. Check all packing and repack if necessary.
4. Move steam pilot valve rods by hand to be sure pilot valve moves easily.
5. Check all connections and fittings to ensure they are tightly in place.

9470.22 STARTING PROCEDURE

To start a reciprocating pump proceed as follows:

1. Oil the pins of the steam valve operating gear and set up on all grease cups.
2. Open the liquid end valves:
   a. Suction.
   b. Discharge.
3. Open the cutout (or root) valves in the:
   a. Exhaust line.
   b. Steam line.
4. Open steam cylinder drains:
   a. Top.
   b. Bottom.
   c. Valve chest.
5. Open exhaust valve at pump.
6. Crack the throttle valve and open slowly so as to admit steam and warm up gradually.
7. Close the steam cylinder drains after the pump makes a few strokes and the steam cylinder is clear of water.
8. Bring the pump up to the proper speed by sufficiently opening the throttle valve. If pump is controlled by a pressure governor, open throttle gradually until governor takes control of pump and then open the throttle valve fully.
9. Adjust the cushioning valves, if fitted, until an adjustment is obtained that permits silent and smooth working of the pump, i.e., sufficient pump speed at the end of the stroke without knocking. After best point of operation is obtained, cushion valve should be set and not changed.

9470.23 STOPPING AND SECURING

To stop and to secure, proceed as follows:

1. Close the throttle valve.
2. Close the exhaust.
3. Open cylinder drains.
   a. Top.
   b. Bottom.
   c. Valve chest.
4. Close the water end suction valve.
5. Close the water end discharge valve.
6. Close the steam and exhaust cutout valves (root valves).
7. After steam cylinder is drained, close the valve chest drains leaving the steam cylinder drain valves open to prevent hydraulic action. (See article 9470.25.10.)

9470.24 FAILURE TO START

If a pump fails to start proceed as follows:

1. Secure the pump. Do not attempt to adjust the tappet collars.
2. Examine the discharge and the exhaust line for closed valves or for a valve disk that has become detached from its stem.
3. If none is found closed, the steam piston may be frozen, particularly if the pump has not been in service for some time. Jack the pump with a bar to determine if there is excessive friction. If so, the trouble probably lies here. (See article 9470.81.)
4. Disconnect the auxiliary valve stem from the operating gear without disturbing the adjustment of the tappet collars. Open the exhaust, suction, and discharge valves and then crack the throttle. Work the auxiliary valve by hand. (The auxiliary valve should work freely by hand.)
5. Should the pump still refuse to start, secure the pump. Remove the steam valve chest cover and examine the main valve to see if it has over-ridden or stuck.
6. If the pump cannot now be started, a complete overhaul of the working parts of the steam end is necessary to stop steam leakage (either in steam piston or valves), the most probable cause of the pump's failure to start.

9470.25 TROUBLES

The main features in the operation of a reciprocating pump, with the principal causes of trouble and the remedies therefor, are enumerated below:

1. **Jerky operation.** Jerky operation on starting is usually caused by a failure to take suction. To correct this proceed as follows:

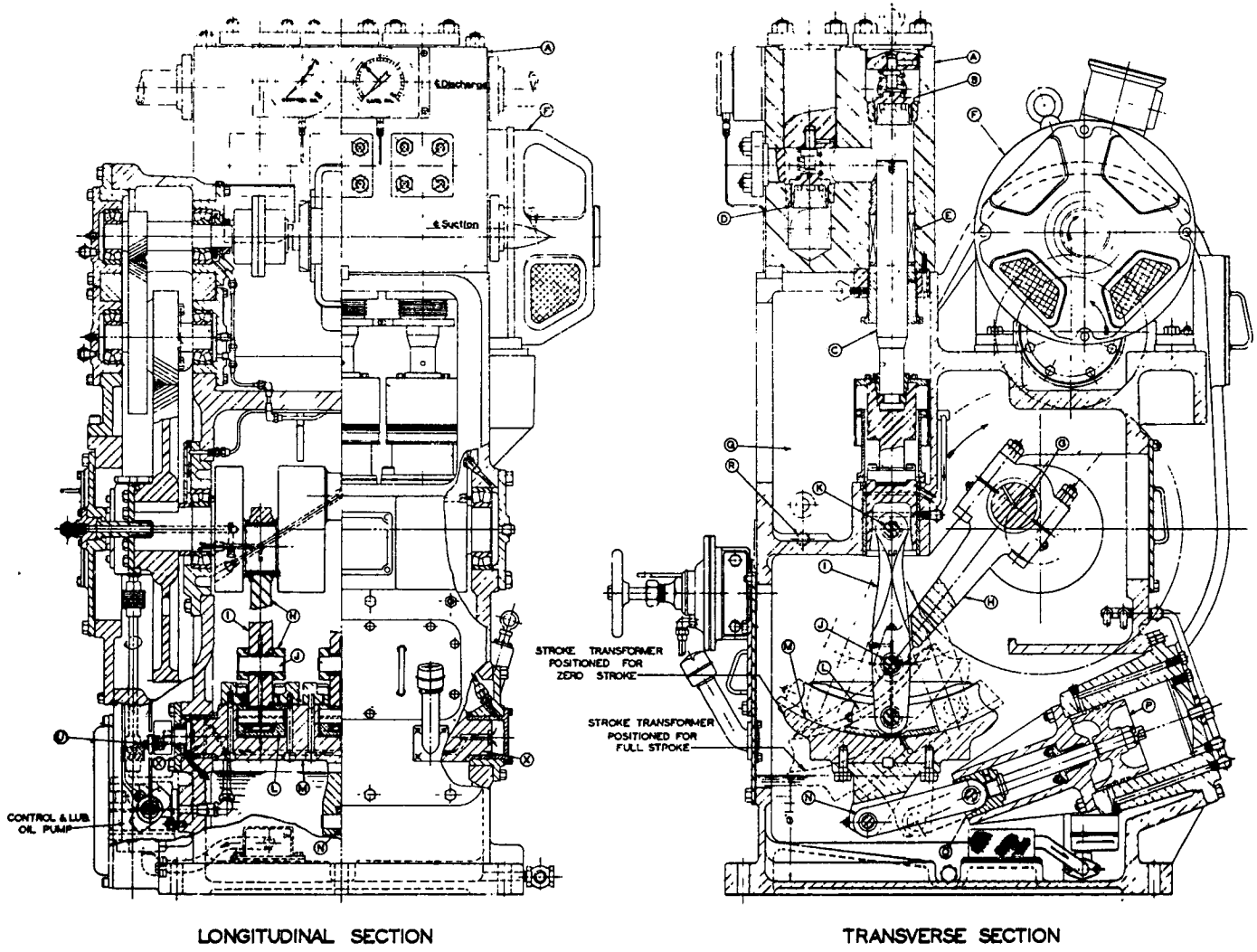
   a. See that all stop or check valves in the suction line are open and that the line is clear of obstructions.

   b. Should a reciprocating feed pump become vapor-bound when taking suction from a variable speed feed booster pump, immediately speed up the latter unit. Unless feed pump regains suction immediately, shift suction to reserve feed tanks until pump is cooled off; turn a hose on the pump cylinder. In the event a standby pump is available, shift to it until the vapor-bound pump has cooled down. Vent valves, if provided, on pump valve chest covers or discharge line will assist the pump in freeing itself of vapor.

   c. Pumps having a suction lift, such as bilge pumps, may require priming before they will take a suction. Salt water pumps can usually be primed from the sea by opening the sea suction valve for a short interval.

   d. Tight piston packing may cause jerky operation.

2. **Loss of discharge pressure.**
   a. Low steam pressure.



LONGITUDINAL SECTION

TRANSVERSE SECTION

Figure 9470-9

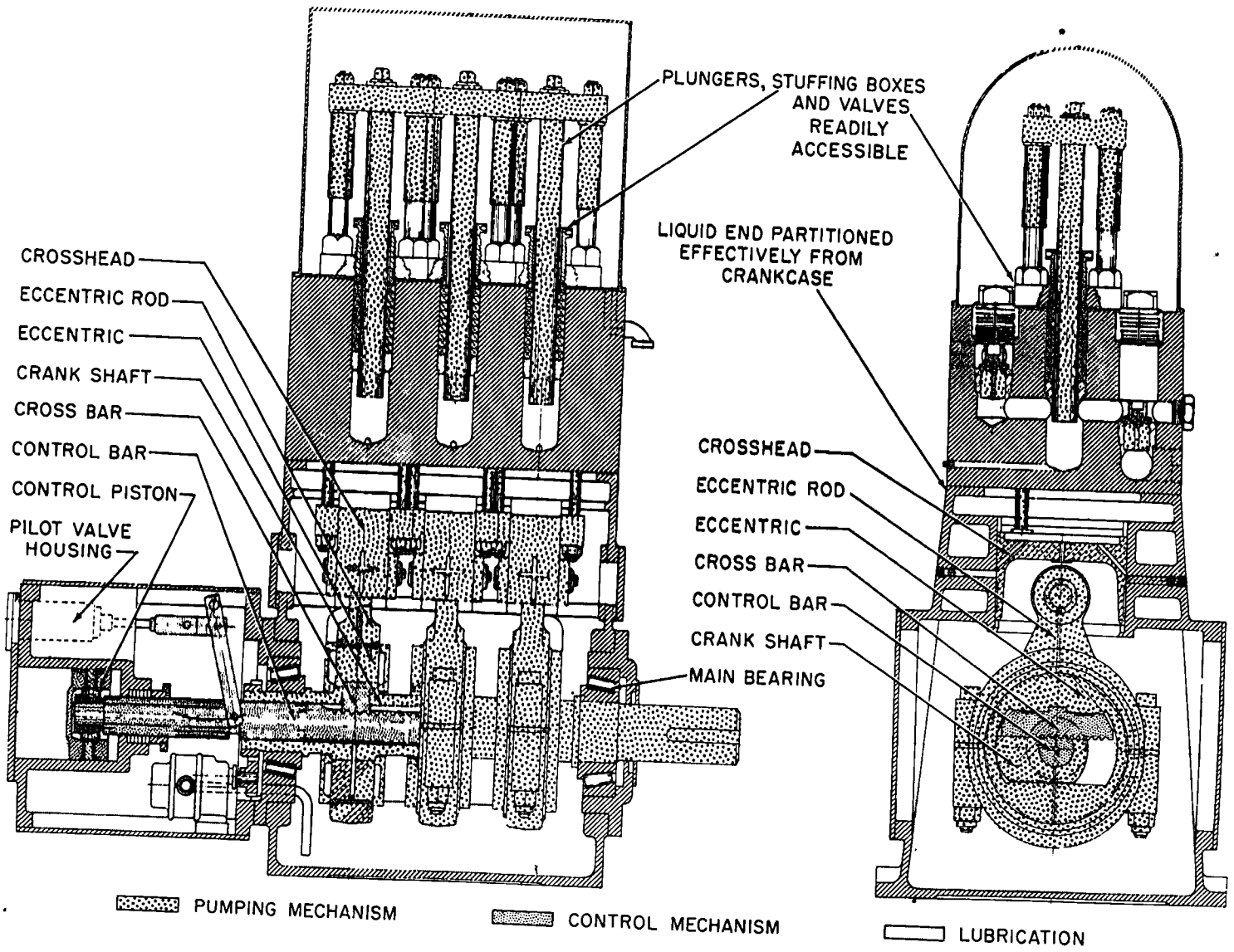


Figure 9479-10

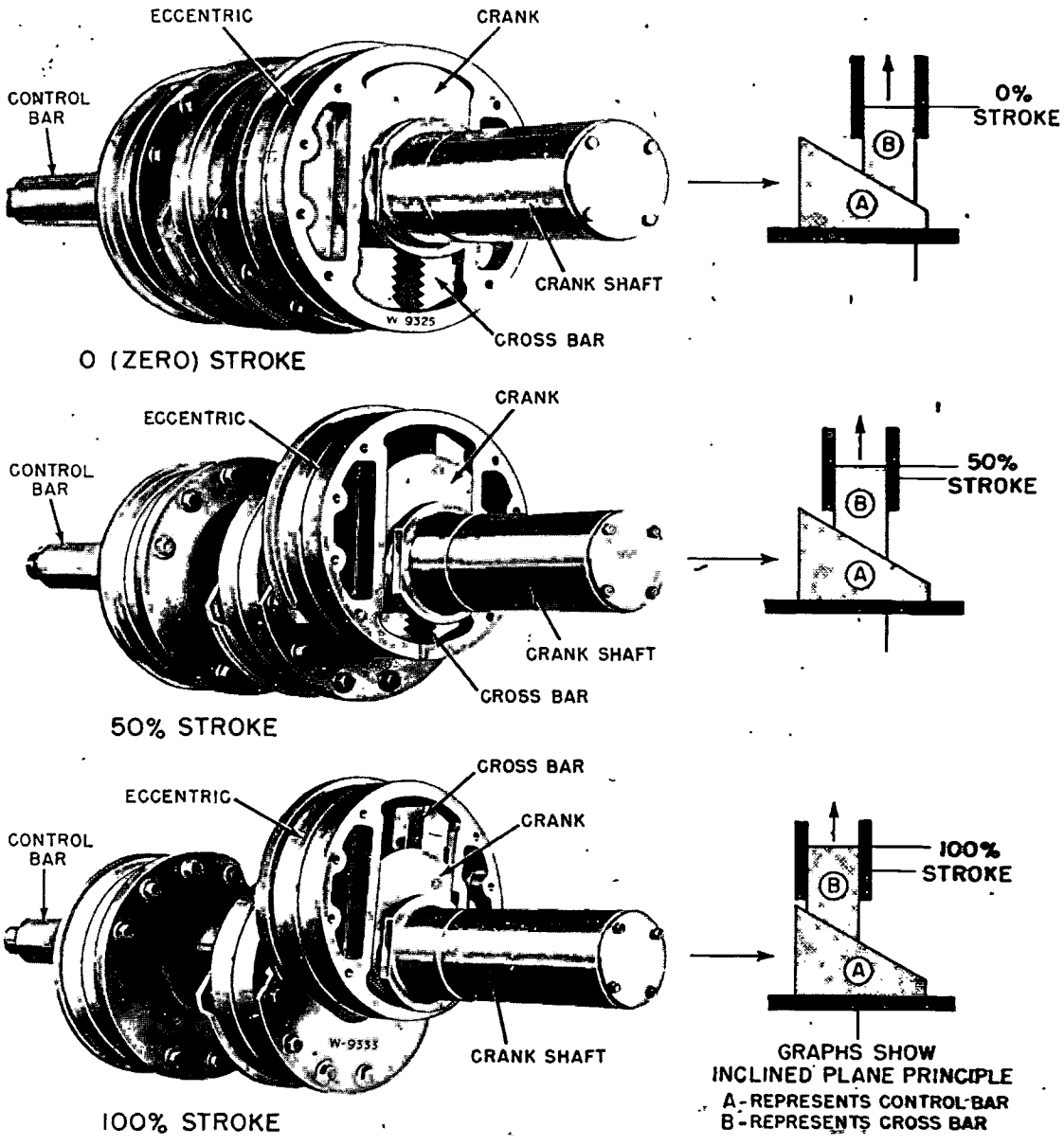


Figure 9470-11

b. High back pressure.

c. Piston rings in steam end worn.

d. Excessive friction loss such as improperly packed stuffing boxes or binding of liquid piston packing.

e. If a pump races without appreciably increasing the discharge pressure, such racing is caused by a leaky piston; leaky, broken, or stuck valve in the water end; or by air being admitted through open or leaky valves in the suction line. Stop the pump as soon as practical in order to ascertain and correct trouble.

f. Should a pump which has been running properly suddenly lose pressure on one stroke, look for a broken valve which should be replaced at once. Great loss in efficiency results from leaky suction, discharge valves, and leaky pistons. Previous experience with a particular type of pump may be taken as a guide in deciding where to look for trouble. Under ordinary conditions, the first investigation should be of the most accessible parts. (See articles 9470.62 and 9470.63 in regard to proper adjustment of valve guards and valves so as to prevent leakage.)

3. **Loss of capacity.** Loss of capacity and failure to develop pressure are closely related troubles, due to the fact that a drop in one will generally be reflected in a drop of the other. Some additional causes of failure to develop required capacity are as follows:

a. Insufficient speed of pump.

b. Suction lift is excessive.

c. Air leakage into pump through stuffing box or leaks in suction line.

d. Entrained vapors in liquid pumped.

4. **Pounding.**

a. Pounding in the liquid end can be caused by improper cushioning, too heavy valves, or a loose piston. This pounding can be stopped by:

(1) Adjusting the steam cushioning valves.

(2) Examine the pistons to see that they are tight on the rod, and the piston rod for lost motion where it is secured to the crosshead.

(3) Look for a loose nest of valves or loose zinc plates, if fitted.

b. If the pump is not fitted with an air chamber on the suction side and pounds, a sniffing valve installed on the suction side usually stops the pounding.

c. An air chamber should be fitted in suction lines when there are high velocities.

d. To stop pounding in the water end of a pump having a considerable suction lift:

(1) Slow down the pump, as this is usually due to water hammer or ram effect in the suction piping.

(2) See that air chamber is properly charged.

5. **Groaning.** Groaning in the water end is generally due to the packing being too tight, but may be the result of misalinement, a broken follower, or a stuck valve. Therefore, the pump should be stopped and examined at once. Failure to investigate usually results in at least a scored cylinder.

6. **Erratic operation.** When a pump operates erratically, sticks in any part of the stroke, or stops frequently, with the throttle valve opened the proper amount, the cause is in the steam end and is probably due to one or more of the following defects:

a. Lost motion in the operating gear due to wear. This should be remedied by rebushing and, if necessary, renewing the pin at the affected part.

b. Leakage of steam by the main or the auxiliary valve due to wear of either. To remedy this, spot in the defective valve on its seat.

c. Leakage of steam by, or sticking of, the valve chest piston. To stop this it is necessary either to rebore the valve chest cylinder and renew or to refit the piston rings.

d. Excessive leakage by the steam piston rings. To prevent this leakage it may be necessary to rebore the steam cylinder or to renew the rings, or both. In some cases where split rings are fitted, the leakage can be partly stopped by taking out the rings and peening them so as to increase the wall pressure. When cylinders are rebored, oversize pistons should be fitted.

e. Small ports and passages in the valve chest stopped up with scale. This frequently happens on new ships as a result of failure to blow out all scale from steam lines before connecting up.

7. **Knocking in steam end.** Knocking in a steam cylinder is indicative of a loose piston assembly or a piston loose on the rod. The pump should be stopped at once and the exact trouble discovered by measurement and examination; then rectified.

8. **Groaning in steam end.** Groaning in the steam cylinder is usually due to steam piston being cocked on the piston rod, to broken rings, or to the cylinders being out of alinement. The trouble, unless immediately rectified, will result in scoring the cylinder walls. If a pump has been idle for a long period, rust some times forms and may cause groaning when the pump is started.

9. **Worn piston rings.**

a. Should the pump stop when going very slowly, it may be due to steam passing worn piston rings. The appearance of a good bearing on the piston ring does not necessarily indicate a good fit. They should be removed from the cylinder so that measurements of both rings and cylinder may be checked for excessive clearance.

b. Should a pump stop or stick, do not jack it with steam turned on; to do so may result in serious damage to it or injury to personnel. (See article 9470.81.)

10. **Hydraulicking.**

a. Water and such liquids as are usually handled by pumps are incompressible. If a pump is fully primed and the discharge outlet closed tight and steam admitted to the steam cylinder, the pump piston and rod would not move if it were not for the slip or leakage from one side of the pump pistons to the other, equalizing the pressure. Under these conditions, on the down stroke or the stroke where the pump rod enters the liquid cylinder, the liquid pressure in the cylinder will build up in the ratio of the area of the steam cylinder to the area of the pump rod. That is termed hydraulic action in pumps.

b. Assuming the size of a steam cylinder as 11-inches diameter and pump rod 2-5/8-inches diameter, and the steam pressure 575 psig, the resulting liquid pressure

$$= \frac{\frac{\pi (11)^2 \times 575}{4}}{\frac{\pi (2.625)^2}{4}} = \frac{95.03 \times 575}{5.42} = 10{,}081 \text{ psig}$$

If such hydraulic action is allowed to occur, it will obviously break the liquid cylinder. This usually occurs when the pump is secured and the steam throttle valve leaks or is not carefully closed.

c. To prevent hydraulic action always open steam cylinder drain valves when pump is shut down and leave them open until pump is again started and cleared of condensation. Check to make sure that relief valve in discharge line functions properly. Make sure that relief valve is located between pump and the discharge shutoff valve in the discharge line.

11. **Vapor lock.** Fuel oil transfer pump suction lines in particular may have vapor pockets, due to the lines having to go over obstructiofis. High viscosity oil in a long line has a high friction loss and it is often necessary to heat the oil to lower its viscosity and the friction loss. This may be accompanied by considerable gasification. With a vacuum on the suction line, this gasification will cause vapor locks wherever pockets exist. This condition also exists when a new oil supply has been taken aboard.

Part 3. Maintenance (where 3-M system is installed, conduct preventive maintenance in accordance with 3-M cards)

9470.31 LUBRICATION

The pins of the valve operating gear should be oiled frequently. Never use oil in the steam or water cylinders, the valve chest, nor on piston rods.

9470.32 JACKING

All pumps should be moved daily and logged.

9470.33 GLAND LEAKAGE

Great care should be exercised in setting up on glands. If the two sides are set up unequally, the gland becomes tilted and scores the rod, and sometimes breaks the gland itself. If a gland is set up too hard, the packing will probably score the rod. If a gland continues to leak after it has been given a few turns on the nuts, the best thing to do is to let it alone until the pump can be shut down. Then an examination should be made to see if the throat bushing is too large, possibly due to the rod having been turned down, or to see if only new packing is needed. Particular attention should be given to packing for cylinders operating at low pressure or below atmospheric pressure to prevent not only loss of vacuum, but the admission of grease into the cylinder and of air into the feed system.

9470.34 STEAM VALVE CHESTS

1. Preventive maintenance of steam valve chests operating under high steam pressures is most important to avoid serious steam cutting and wear. Routine inspections will disclose early signs of leakage or wear and need for corrective action. Careful attention to lapping out these early cuts will ensure a tight metal-to-metal fit of the valve chest and steam cylinder.

2. Attention to condition of piston rings on the main and auxiliary valve with timely replacement will preclude broken rings and steam cutting in the grooves.

3. Prior to making up the metal-to-metal joint, it should be blued and tested for good overall contact. If it is not a good metal-to-metal fit of if scores or leakage are evident, remove the studs from the steam cylinder and lap the steam chest in against the cylinder seat.

4. Strict cleanliness must be observed in making up the joint so that no particles of dirt or scale will prevent a good metal-to-metal contact of the ground faces. A thin film of copaltite should be applied. The holding-down bolts should be taken up gradually and alternately in a diagonal sequence. After steam is applied and the pump is in operation, the holding-down bolts should be taken up again.

5. Do not install gaskets on steam chest joint. Such a gasket tends to blow out around the steam passage lands, thus aggravating the cutting action of the steam.

6. When continued lapping of valve chest and steam cylinder surfaces have worn them down to the extent that they must be built up to original dimensions, then both should be overlayed and brought back to design dimension. When this is necessary, follow the instructions in article 9470.75.

9470.35 ADJUSTMENT OF STROKE

1. Pumps should be made to run with the full length of stroke which will ensure that the piston travels a little beyond the counterbore.

2. Full stroke is that which is stamped on the nameplate. (See article 9470.2.2.) A stroke indicator, consisting of a sheet metal pointer secured to the piston rod crosshead and two marks on one cylinder tie rod, is usually provided. The upper mark lines up with the pointer on the crosshead when the pump is at the upper end of its stroke, while the lower mark indicates the proper position of pointer with pump at the lower end of its stroke. If such stroke indicator is not already fitted, it should be installed by the ship's force, as it materially assists in the proper setting of the valve gear, and permits a ready checkup of pump stroke. This ensures steam economy and better operation of the pump. When full stroke cannot be obtained, it invariably means that something is wrong with the adjustments of the pump. A short stroke results in incomplete cushioning and the formation of shoulders in the cylinders and valve chests with resultant breakage of rings and followers. A long stroke is usually indicated by a heavy metallic knock in the steam cylinder and should be immediately corrected. It should not be necessary to keep moving the tappet collars continually while the pump is running; if found necessary to do so, something is wrong with the pump and it should be dismantled and the interior parts examined. Do not alter the settings of the tappet collars without orders from the engineer officer.

3. The proper valve setting must be carefully determined to make a pump take full stroke. Refer to manufacturer's drawings and instruction books for detailed information as to the method to be followed for specific makes and types of valve gear. The following general method is included as a guide when more detailed information is not available: Place piston and auxiliary valve on the center or half stroke. Then move each collar so that they will be about one-half the width of the steam port away from the tappet. Start the pump. If the stroke is too short, the collars should be screwed farther apart. Care should be taken to move both collars the same amount, otherwise stroke will be longer on one end than on the other. When the final adjustment has been made, lock the collars securely in place.

4. a. To adjust the steam valves of a duplex pump, place one piston on its top striking point and, by removing the steam chest cover of the valve chest of the other piston, adjust and secure the valve of the other piston to give an excess on one-eighth inch of full port opening at the top end. To set the other valve, repeat the operation with the other piston.

b. Run the pump slowly with the throttle cracked, against little or no pressure, and with the cushioning valves wide open if fitted. The pistons should be striking on the cylinder heads; if they do not, there is some undue friction, possibly due to tight piston rod and plunger packing. Close down on the cushioning valves until the pump is running at full stroke of both steam pistons with smoothness of reversal and no striking. Should it be impossible to obtain this smoothness of reversal, it may be necessary to alter slightly the adjustment of the valve operating collars.

c. To shorten the length and prevent striking of the piston against the cylinder heads, set the collars close together; to lengthen the stroke, set the collars farther apart. Care should be taken to move both collars the same amount, otherwise stroke will be longer on one end than on the other.

5. Some pumps may, due to design characteristics or wear, override and pound at full speed after having been adjusted to give full stroke at low speed. The only remedy for this is to have different valve adjustments for different speeds.

6. After a pump runs on short stroke for a considerable length of time, difficulty may be encountered in making the pump take a full stroke, due to shoulders having been worn in the steam valve seat and pump cylinder lining.

These shoulders will have to be removed before full stroke can be obtained.

9470.36 PROTECTION AGAINST CORROSION

On salt water pumps having cast iron bodies and parts in contact with salt water, special care shall be taken to guard against corrosion. These parts shall be examined every six months, scaled, and wire brushed insofar as practical. Reduction of corrosion can be effected by fitting zinc plates or rings on water end valve chest covers and on cylinder covers.

9470.37 EXAMINATION OF WATER VALVES

For satisfactory and economical running of pumps, it is essential that the valves should be absolutely tight. The liquid end valves in all pumps should be examined quarterly and all foreign matter on valves, valve stems, and valve springs removed.

9470.38 TESTS AND INSPECTIONS (RECIPROCATING PUMPS)

1. **Weekly.** Operate all pumps by steam or power. If power is not available, move by hand.
2. **Quarterly.** Check settings of relief valves.
3. **Annually. (See also 9470.52, para. 2)**
   a. Inspect liquid end, valves, valve stems, and springs.
   b. Inspect steam valve gear for wear.

NOTE: These tests and inspections are the minimum necessary to give adequate assurance of safe and reliable operation of equipment. Indications of low discharge pressure or other manifestations of improper operation should indicate more frequent or extensive tests and inspections.

Part 4. Repairs

9470.51 GENERAL

If a pump does not function properly and is in need of repair, do not dismantle the steam end until a thorough investigation shows that the trouble is not at the water end. Most pump troubles are due to fouled water cylinders, worn valves, or conditions in the pipe connections external to the pump.

9470.52 DRAWINGS AND DATA

1. When repairing or making an examination of a pump, it is essential that assembly and detail drawings and available dimensional data thereto be at hand. Frequently after overhaul, important dimensions become altered, such as the width of and distance between steam ports, the length of rods and steam valves, and the diameter of pistons. This may cause poor operation which will continue in spite of other major repairs unless the real cause of trouble is recognized and rectified.

2. Whenever reciprocating pumps are opened up for repairs (at least annually), micrometer measurements shall be taken of the cylinders and valve chest on the fore and aft and athwartship diameters at the top, middle, and bottom, and the results recorded on Form OPNAV 4700-2G (NAVMAT Notice 4700) card with an accompanying diagrammatic sketch showing measurements obtained and the date on which they were made.

9470.53 SCORED WATER CYLINDER

A scored water cylinder need not necessarily be rebored or renewed. Scores should be smoothed up by stoning to reduce leakage and wear on the packing. When the scoring is extensive and the packing rapidly wears out, reboring or renewal of the liner should then be considered.

9470.54 LOOSE PISTONS

1. Steam and water pistons frequently work loose on the rods. This fault may be due to poor fitting so that the shoulder on the rod brings up against the piston without giving a proper bearing surface for the tapered part of the rod. The piston should fit within 1/32 inch to 1/8 inch of the shoulder when set up hand tight, depending on the taper of the rod, and should then be forced tightly against the shoulder by the securing nut. A small amount of foreign matter on the tapered part of the rod will prevent the piston from being brought home firmly. If a piston which has given trouble is properly refitted to the rod, the trouble will usually disappear.

2. If a jamb nut or split pin is not fitted, one should be installed.

9470.55 TEST FOR TIGHTNESS

The following method of testing the tightness of the steam piston in the cylinder has been used with success:

1. Remove the cylinder head. Shore the piston to prevent forward or upward motion.

2. Connect a steam hose to the lower drain cock and gradually raise the pressure to the working pressure of the pump or the hose, whichever is less. If the rings are not tight, steam will leak past them. It must be remembered that this test shows defects for only that portion of the cylinder occupied by the piston. If measurements show that there is a great difference in size in various parts of the cylinder, the test should be repeated several times with the piston in a different part of the cylinder during each test.

9470.56 PISTON TOO SMALL

On reboring a water cylinder, the piston and follower may be given so much clearance as to require renewal. As a temporary substitute, until a new piston and follower can be obtained, one of the following procedures may be employed:

1. Install upper and lower rings of metal packing and insert soft packing between them.

2. The piston and follower may be turned down, threaded, and a ring with same thread screwed tightly on. The outside diameter is then machined to fit the cylinder.

9470.57 BUILDING UP PISTONS

Pistons may be built up by flowing on metal by the oxyacetylene method, electric welding, or metal spray. The piston is then machined to proper fit.

9470.58 FITTING SOFT PACKING

When packing a pump with Tuck's, flax, or other soft packing, soak the packing in hot water approximately 12 hours before fitting and installing. Should an urgent demand for the pump make this impractical, the packing should be put in to fit loosely and allow for swelling. A

failure to do this will surely cause the pump to groan or result in a scored cylinder.

9470.59 LAMINATED PHENOLIC PACKING

Laminated phenolic packing type rings are increasing in usage, particularly for boiler feed service. The Naval Ship Systems Command is conducting laboratory and service tests on various types and makes to determine their suitability and to develop data as to their proper application and installation. The use of laminated phenolic rings as a substitute for soft packing may be authorized by NAVSHIPS upon request, provided the design of the piston is satisfactory for or can be altered to accomodate the phenolic type rings.

9470.60 METAL RINGS

Metal rings are sometimes very narrow and cut the cylinder. If wider rings are fitted, the trouble generally stops. Sharp edges of rings should be rounded with a file. The use of white metal rings is considered obsolete and unsatisfactory.

9470.61 BREAKING OF FOLLOWERS

1. Breakage of followers and bolts may be due to misalinement. Another cause may be screwed plugs in the piston working loose and coming adrift. Whenever a piston is removed for examination or repairs, check for tightness of plugs. Plugs should be prick-punched to prevent their backing out.

2. However, if this trouble in general in any one set of pumps on board ship, it may be that the followers are too weak, in which case the Naval Ships Systems Command should be advised so that the design may be improved and alterations authorized.

3. In pumps where the follower is too small in diameter, soft packing may roll up between it and the cylinder causing the follower to jam and break.

9470.62 VALVES IN WATER CYLINDER

1. For satisfactory and economical running of pumps, it is essential that the valves in the water end be absolutely tight. Cast disc valves may be faced off in a lathe and then ground in on their seats by a simple device which consists of a rod of sufficient length, slotted for a piece of metal which fits into a slot across the top of the valve. An ordinary bit stock can be used to do the grinding.

2. It is sometimes desirable to take a cut off the valve seat without removing it. A simple cutter for this can be made with an extension for a bit stock similar to the grinding device. When flat valves are fitted, the seats may be trued up by using a small surface plate, and spotting in the section on the surface plate.

3. At each examination try all sheet metal valve discs with a straight edge to see if they are true. Solid (cast) valves, if scored or warped, should be trued up in a lathe. Kinghorn valve (thin metal) discs, figure 9470-9, if dished or warped should be discarded and new ones installed, as they are liable to break after having been dished. The life of rubber valves can sometimes be prolonged by trimming and turning the valves and by inserting brass backings.

4. See that valve springs have the proper tension; this should be just great enough to ensure a quick closing of the valve and it should be possible to lift the valve easily by hand. See that the springs are well secured by split pins. Adjust the valves to give the proper lift; lift should be such that the circumferential opening is slightly greater than the clear opening through the seat. The lift ought never to be more than one-fourth of the diameter of the opening.

5. Keep valves clean; a light mineral oil makes a good cleanser and a lye or soda solution is good for removing caked or gummed oil from valves.

6. New valves may differ in thickness from the old, thus giving too much or too little lift; too little lift will usually cause the valve to squeal.

7. In some pumps the valve seats are secured only by a taper fit. In such cases the valve seats should be forced home by a jack resting on the end of a reseater which in turn rests on the face of the valve seat. Should the seat work loose, slightly peen over the edge of the metal. In pumps that have the valve seats screwed into the pump, diaphragm, they should always be screwed in with white lead, otherwise it will be almost impossible to get them out.

8. Several valves are sometimes secured in one plate. The plate is secured in the water valve chamber with a ground joint which is difficult to make tight. If it is not tight, water will leak under the joint and score it. This joint can be reseated with ship's valve reseating machine or by being ground in with a valve grinding mixture. In difficult cases a copper gasket may give better results.

9470.63 ASSEMBLING VALVES

Great care should be exercised in assembling pumps after overhaul; otherwise, the entire work on valves, etc., may be ·t. Valves, seats, stems, and springs should be marked before removal and replaced in the same place from which they were removed.

9470.64 SNIFTER VALVE

When it is thought that a snifter valve is the cause of faulty pumps performance, it should be borne in mind that the valve is simply a check valve with an adjustable amount of opening, and its functioning is usually to prevent overloading and to permit the entrance of a certain amount of air at each stroke and thus do away with the necessity of charging the air chamber with high pressure air. A globe valve should be installed between the check valve and the pump so that the amount of air can be regulated.

9470.65 RELIEF LINE

If the relief line from a feed pump is piped back to the suction line to a point close to the pump, water may be blown away from the suction when the relief valve lifts, causing the pump to become vaporbound. This trouble can be stopped by piping the relief line to the feed tank.

9470.66 BLOWING GASKETS

1. For all cold water pumps a sheet packing of rubber with cloth insertion is suitable for the water end bonnet. For the steam end of all pumps and water ends of hot-water pumps, compressed asbestos packing is used for bonnet gaskets for steam pressures up to 300 psi. Above this pressure metal-to-metal joints should be used, or gaskets of soft iron or steel annealed dead soft. Maximum hardness of 80 Brinell may be used. When gasket blowouts occur, the trouble can usually be eliminated by turning a shoulder on the head and counterboring the cylinder, so as to make a male and

female joint; this shoulder will prevent the gasket from blowing out.

2. On the water cylinder the head with the stuffing box gives the most trouble by blowing gaskets, on account of an alternate increase and decrease of pressure on the gasket, due to the motion of the rod. This will be aggravated if the stuffing box is unduly tight. A very reliable joint for this end can be made by fitting the shoulder as above described and then using a copper gasket cut out in one piece and put on by disconnecting the rod at the crosshead.

3. In order to keep gaskets from blowing out through the forcing bolt holes, the forcing bolts should be kept in place.

9470.67 SCORES IN STEAM CYLINDER

Relatively small scores in a steam cylinder necessitate a reboring on account of the cutting action of steam. Leakage past the steam piston is readily located by the dullness and discoloration of the cylinder walls. Having once started, through the medium of a few scores, steam will gradually cut away cylinder walls until leakage by the piston becomes so excessive as to cause faulty operation of the pump. The spring in steam piston rings will take up some of the wear in the steam cylinder, thereby permitting some wearing away before reboring the cylinder is necessary.

9470.68 THE ALINEMENT OF PUMPS

1. Pumps must be kept in line, as failure to keep proper alinement is one of the greatest sources of trouble with pumps aboard ship and is one of the most serious defects. Pumps secured to a bulkhead are more likely to get out of alinement than those with an independent base and setting. A pump may have been well alined in the shop and then pulled out of line when it is bolted to the bulkhead, or, after installation, the bulkhead may have distorted sufficently to cause misalinement. A pump run in this condition usually scores the rod and cylinders and breaks follower and bolts. The alinement of pumps should be tested occasionally by removing the pistons and rods and running a line through the cylinders; especially should this be done during the first year of a new ship in commission and also if a pump is giving trouble by scoring the rod or cylinders or breaking followers.

2. It is found in some cases that when steam cylinder foundation pad bolts are slacked off, the cylinder pad pulls away from the foundation as much as one-half inch, indicating settling of foundations and bulkheads. This is corrected by fitting shims between foundation and pump.

3. An easy method for a rough check of alinement is to pull the steam and liquid end rod packing, and check clearances at four points 90 degrees apart between piston rods and cylinder heads as soon as practical. Alinement should be checked at three positions—top, center, and bottom of the stroke. If throat bushings are not worn (out of round) and clearances are not uniform, realinement of pump should be undertaken. Misalinement may also be due to lack of provision for expansion of the pump with respect to foundations. This can be corrected by longating the foundation bolt holes in the steam cylinder foundation pad, or using foundation bolts one-eighth inch smaller in diameter than the holes.

4. A frequent cause of misalinement (and often not recognized) is the piping. The piping should line up naturally with the pump connections and should not be forced into position. Provision must be made in steam and exhaust lines to take care of expansion. All piping should be supported independently of the pump. Incorrectly installed piping puts a strain on the pump and forces it out of alinement. Special care should be taken in the installation of suction piping to prevent vapor pockets, particularly on fuel oil pumps.

9470.69 RUNNING A LINE

To run a line through the cylinders to facilitate the work of realinement, fasten one end of the line on a finger piece secured to the bottom of the water cylinder and the other on a temporary beam rigged above the steam cylinder; then center the line at the bottom and top of the water cylinder so that it becomes the axis of this cylinder. The steam cylinder can now be moved about and centered on the line without disturbing the centering of the line on the water cylinder.

9470,70 FITTING METAL RINGS

1. Troubles in the steam cylinder result chiefly from the rings. Leakage of steam past a ring dulls the rubbing surface of a ring which, when tight, has a highly polished glazed finish. Diagonally cut split rings seem to give better service than overlapping rings. The latter generally break at the corner after being in service a short time. However, this trouble can be reduced by rounding the corners of the shoulders.

2. The ideal piston ring would be one which would prevent completely the leakage of steam without exerting lateral pressure against the cylinder wall. This, however, is impractical so split rings are used. The method of fitting them to the cylinder varies with the different designs of pumps. Ordinarily the designs of rings shown on approved drawings or manufacturer's instruction book of the pump should be followed in making up and fitting new rings. In the absence of drawings or with continued failure of rings, the practice of the home naval shipyard or repair ship should be followed.

3. The following procedure is one which will usually be found to give satisfactory results. In order that the ring may be under tension and still remain in close contact with the cylinder wall, even after a little wear, and at the same time not allow excessive leakage through the joint, a ring should be turned before splitting to a diameter that is 0.010 of an inch per inch of diameter larger than the cylinder bore. The ring is then diagonally split and an amount taken out which is approximately twice the amount of difference between the diameter of the rough turned ring and the cylinder bore. A liner consisting of a piece of chart paper for the larger size and thinner paper for the smaller size pumps is now placed in the gap made by splitting. The ring is secured on the face plate of a lathe and turned to the exact diameter.

9470.71 TIGHTNESS OF STEAM VALVES

Flat steam valves should be kept true by scraping and the packing rings of piston valves kept free in the grooves. When scraping flat steam slide valves, the strokes of the scraper should cross, so that the scraper will not chatter on the narrow bridges between the ports, and thus cause steam leakage when the valve is in operation. It is highly

important that all the steam valves be kept tight, for, if the auxiliary valve leaks, the main valve will become steam bound; that is, the steam pressures in the various passages and ports of the auxiliary valve chest and in the main valve chest cylinders will equilize and the pump will stop.

9470.72 TO CHECK VALVE MEASUREMENTS

As a result of the frequent spotting in of the main and auxiliary valves, the relative size and arrangement of port openings may change owing to irregular coring. An easy method to check the accuracy of the valve action is to cut paper patterns of the valve and valve seat faces by laying the paper on the valve and peening with a hammer. Sliding the pattern of the valve over that of the seat will show the exact laps, leads, and port openings, which can then be checked with the drawings.

9470.73 VALVE CHEST STEAM CYLINDER

The same care must be exercised in fitting the valve chest steam cylinder, cylinder rings, and piston as is used in fitting the steam piston and cylinder.

9470.74 CARE OF MECHANISM

It is most important that all wear be kept out of the steam valve operating mechanism. Failure to do this will cause the pump to operate in a faulty manner and perhaps to stick. Rebush and renew the pins as often as necessary. If the wear of the bushings occurs rapidly despite careful lubrication, the holes may be bushed with tool steel bushings.

9470.75 REPAIRS TO VALVE CHESTS, LINERS, AND PISTON VALVES

1. When piston valve chest joints and lands and other parts are badly damaged by wear and steam cutting, the following repair procedures are authorized. Refer to figure 9470-12 for identification of the areas to be repaired.

2. Joint face "A" between steam chest and cylinder may be built up by overlaying with one of the following electrodes in accordance with Specification MIL-R-17131. Both surfaces should be brought back to their design dimensions.

3. Main liner chest lands "B" may be built up with 25 percent chrome 20 percent nickel corrosion resisting (austenitic type) steel welding electrode.

4. Liners should be replaced rather than subjected to extensive repair. However, if spare liners are not available, liner may be chrome plated on the outside area "C". In such event liners should be shrunk into place. Pressing chrome plated liners into the chest will result in galling. Chrome plating the inside or wearing surface of liners area "D" is not authorized as a repair as this would do away with the graphitic lubrication provided by the original special alloy iron liners.

5. Main and auxiliary valves should be replaced when worn or steam cut. Building up ring grooves "E" with chrominum cobalt composition is authorized if replacement valves cannot be obtained in time to effect repair. The original valve material has a hardness of approximately 200 Brinell compared to approximately 500 Brinell for the liner. This is designed so that wear will be taken on the pistons rather than on the more expensive liners. Thus to overlay the ring grooves with a harder material than the liner will result in rapid wear of the liner and is to be avoided.

6. The end lands of main piston valves may be found to have insufficient clearance with the liner. The clearance on diameter between end lands "F" and the liner should be between .002 and .0025 inches per inch of diameter. Center lands should have approximately half of this clearance.

7. Replacement piston rings "G" should be of cast iron as originally installed in order to obtain the required finish on working faces.

8. Tail rods "H" should be made of K monel metal when replaced in order to avoid corrosion and the danger of rust working loose in the valve chest. Chrome plating of the original carbon steel rods is satisfactory as a repair.

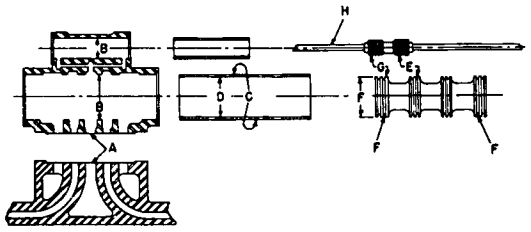


Figure 9470-12

Part 5. Safety Precautions

9470.81 SAFETY PRECAUTIONS

1. Never use a jacking bar to start a pump while the steam valve to the pump is open.

2. Boiler feed pumps shall not be used for purposes other than those connected with the service of the boilers or use of feed water, except in an emergency.

3. Before opening a steam cylinder or steam valve gear, see that drains are open and that steam and exhaust root valves are wired closed.

4. Before opening the water cylinder or valve chest of a pump handling water at a temperature in excess of 120° F., see that suction and discharge valves are wired closed, and cylinder and valve chests are drained.

5. Always open steam cylinder drain valves when pump is shut down and leave them open until the pump is again started and cleared of condensation.

SECTION II. CENTRIFUGAL PUMPS

Part 1. General

9470.101 INTRODUCTION

Centrifugal and axial flow pumps are now largely used for all nonviscous liquid services on naval ships. They have largely supplanted direct-acting steam pumps for such services. Advantages of centrifugal pumps include simplicity, compactness, weight saving, and adaptability to high-speed prime mover. However, centrifugal pumps have the disadvantages of lack of suction power and must be primed to operate on a suction lift. They are also sensitive to variations in head and speed. They are usually designed for

a specific set of operating conditions and may give unsatisfactory performance when subjected to operation at great variance with the design rating. Hence it is important that the principle of operation, the suitable application, and limitations of the various types be thoroughly understood.

9470.102 DESCRIPTION

1. A centrifugal pump consists essentially of a hydraulic machine which imparts energy to the liquid through centrifugal force. This force is produced by the rotation of an impeller at high speed within a casing. Liquid is led to the "eye" or center of the impeller and is thrown to the outer periphery of the impeller by the centrifugal force imparted by the impeller vanes. The liquid leaves the impeller with high velocity and kinetic energy and is collected by the casing volute or diffusion vanes. These and the diffusing discharge nozzle are carefully designed to convert the high velocity kinetic energy to low velocity potential energy or pressure at the discharge flange of the pump.

2. The axial flow or propeller pump has a flow parallel to the shaft. The liquid has velocity imparted by the action of propeller vanes and this velocity is converted to pressure by a diffusion casing and diffusion vanes which straighten the flow and reduce its velocity. Correctly, axial flow or propeller pumps are not centrifugal pumps, but are not considered as a separate class because they represent the high capacity, low head extreme type in the whole pump family.

3. Intermediate types between the pure radial or straight vane centrifugal impeller and the axial flow include the "francis vane" and "mixed flow" pumps.

4. Figures 9470-13 and 9470-14 show diagrammatic section through volute and diffusion type centrifugal pumps respectively.

9470.103 CLASSIFICATION AND TYPES

There are many methods of classifying centrifugal pumps, including:

1. **Service application.** (See article 9470.104.)
2. Number of stages.
   a. Single stage.
   b. Two stage.
   c. Multistage.
3. Shaft position.
   a. Horizontal.
   b. Vertical.
   c. Inclined.

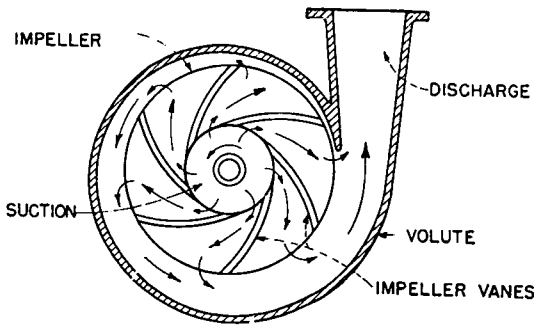


Figure 9470-13

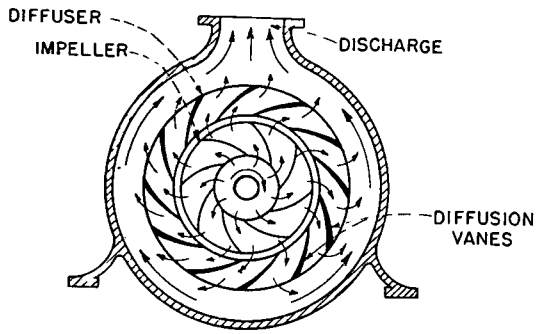


Figure 9470-14

4. Type of impeller.
   a. Single suction or double suction.
   b. Open, semiclosed, or closed.
   c. Radial, francis, mixed flow, or axial flow (propeller).
5. Type of casing.
   a. Volute, diffuser, or turbine.
   b. Solid, split, or barrel.
6. **Nozzle arrangement.** Side, end, top, or bottom and identified to both suction and discharge nozzles.
7. **Materials.** All iron, bronze-fitted, all-bronze, steel, stainless steel, titanium and other special alloys.

NOTE: The first two material classes are seldom used in naval shipboard service due to their inherent lack of shockproofness and accelerated corrosion when handling sea water.

8. **Type of drive.** Flexible coupled, rigid coupled, close coupled, gear or belt driven.
9. Other classifications such as special construction, direction of rotation (or reversible) kind or power drive, etc.
10. Figures 9470-15, 9470-16, 9470-17, 9470-18, 9470-19, and 9470-20 illustrate the several types of impellers mentioned herein.

9470.104 SERVICE CLASSIFICATION

The various pumps used in naval ships have requirement and problems of operation and installation peculiar to each application. The more distinctive types are listed below according to application. As types of naval ships vary greatly, not all of these services will be found on all ships; also special machinery plants may require pumps not listed herein:

1. Boiler feed pumps.
   a. Main.
   b. Auxiliary.
   c. Port.
   d. Cruising.
2. Feed booster pumps.
   a. Main.
   b. Auxiliary.
   c. Cruising.
3. Condensate pumps.
   a. Main.
   b. Cruising.
   c. Auxiliary (or dynamo).
   d. Distiller.

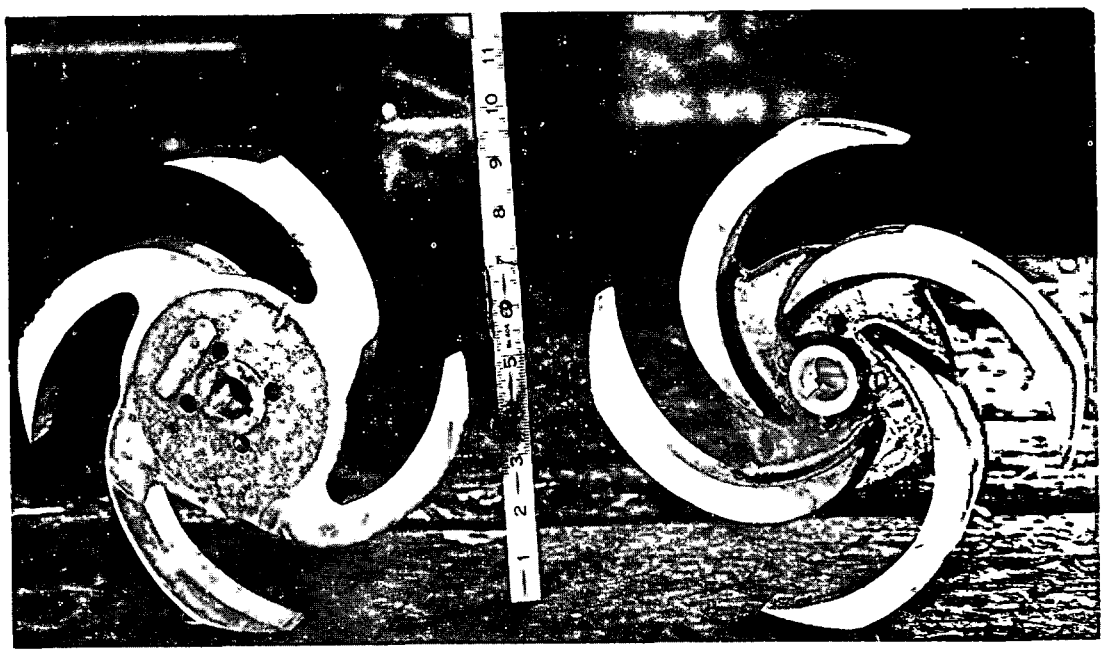


Figure 9470-15

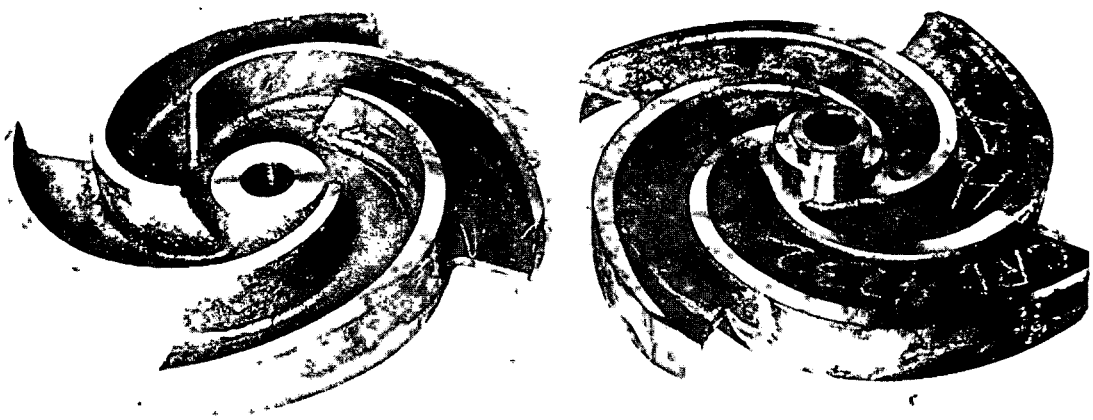


Figure 9470-16

4. Circulating.
   a. Main.
   b. Auxiliary (or dynamo).
5. Fire pumps–including combination services such as fire and flushing, fire and emergency bilge.
6. Miscellaneous circulating and supply pumps
   a. Distiller condenser circulating.
   b. Refrigeration condenser circulating.
   c. Diesel engine cooling water.
   d. Auxiliary machinery cooling water.
   e. General service.
7. Fresh water pumps.
   a. Ship's fresh water.
   b. Fresh water pressure system.
8. Distilling plant pumps.
   a. Distiller circulating.
   b. Distiller condensate (distillate).
   c. Evaporator feed.
   d. Brine overboard.
   e. Distiller fresh water.
   f. Evaporator first effect coil drain.
9. Gasoline pumps.
   a. Main.
   b. Airplane fueling and defueling.
10. Lubricating oil.
    a. Main lubricating oil.
    b. Main lubricating oil standby.
11. Cargo.

a. Fuel oil.
b. Gasoline.

12. Portable pumps.
    a. Electric submersible.
    b. Gasoline engine driven.

9470.105 GENERAL CHARACTERISTICS

1. Unlike positive displacement pumps, a centrifugal pump at constant speed will deliver liquid at any capacity from zero to a maximum capacity which is limited by size of the pump, the suction condition, and its design factors. Over its range of capacity, the pressure, efficiency, and power required vary according to relationships inherent in the particular design. The relation of capacity, total head (pressure developed) efficiency, and horsepower required is customarily expressed by means of a characteristic curve. A characteristic curve generally contains the following three curves:

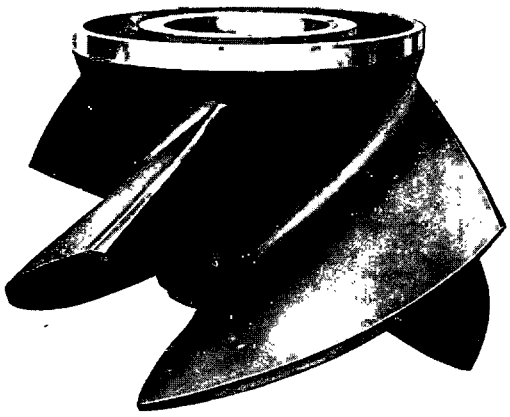


Figure 9470-19

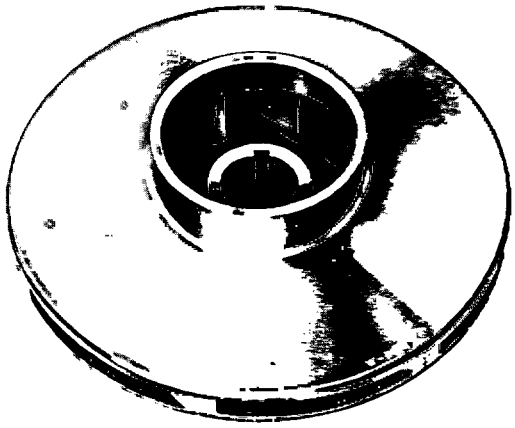


Figure 9470-17

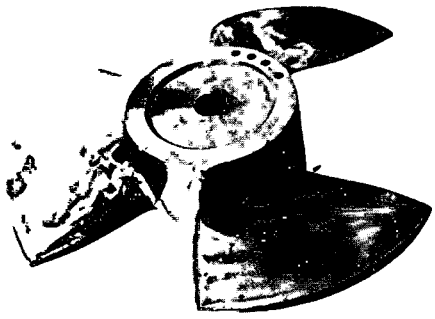


Figure 9470-20

a. Head plotted against capacity.
b. Efficiency vs. capacity.
c. Brake horsepower vs. capacity.

All three curves are at constant rated speed. A typical characteristic curve is shown in figure 9470-21.

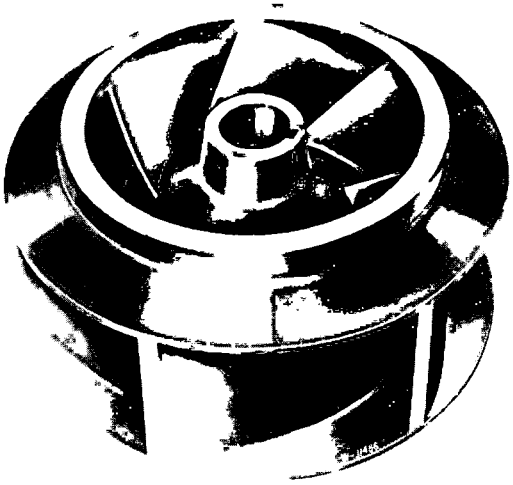


Figure 9470-18

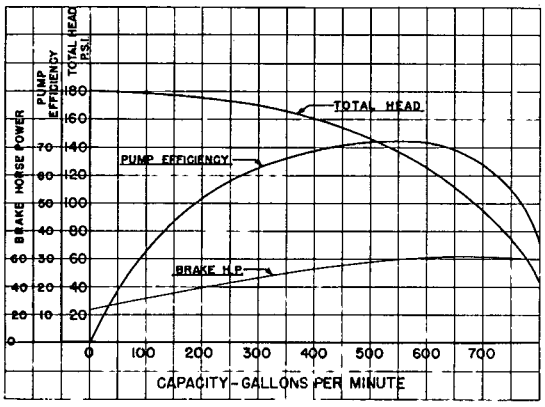


Figure 9470-21

2. In all centrifugal pumps the following laws of centrifugal apparatus hold true within practical operating limits of the pumps. For a given pump:

a. The capacity varies directly as the speed.

b. The total head varies as the square of the speed.

c. The horsepower varies as the cube of the speed.

3. It does not necessarily follow that when increased capacity is desired that it will require an increase in speed in direct ratio. This will depend on the pressure required. The performance acceptance test curves of the individual pumps should be consulted to determine the effect of changes in pump speed and pressure.

4. The following formula will be found useful for determining the approximate total head a centrifugal pump will develop:

a. $H = f\left(\frac{DN}{1840}\right)^2$

when H = total head in feet of liquid.
D = impeller diameter, inches.
N = pump revolutions per minute.
f = a speed constant.

f at shut off varies from 1 to 1.15

f at maximum efficiency varies from 1.10 to 0.75 and an average of 0.95 is accurate enough.

b. To convert H in feet to pressure psi, divide H by 2.31 for cool fresh water, or 2.24 for sea water.

c. For a multistage pump having equal impeller diameter, determine H for one stage and multiply by the number of stages, otherwise determine H for each stage and add the values of H.

d. Since the suction pressure is seldom zero, the discharge pressure to be expected should be computed from the total pressure by adding the positive suction pressure or by subtracting the suction lift or vacuum.

5. The power required to drive any centrifugal pump is determined as follows:

$$WHP = \frac{GPM \times H \times sg}{3960}$$

where WHP = water horsepower output.
GPM = capacity in gallons per minute.
H = total head in feet of liquid.
sg = specific gravity.

or $WHP = \frac{GPM \times P}{1714}$

when P = total pressure in psi the

$$BHP\ required = \frac{WHP}{pump\ efficiency}$$

9470.106 SYSTEM CURVES

A system curve is a plot of the pressure characteristics of the system against variations in flow. For any given capacity the system pressure is the summation of friction losses, pressure in receiver minus pressure at the source, plus static pressure. Typical system curves for a condenser circulating pump (in which the losses are practically all friction) for a main condensate pump, and for a boiler feed pump are shown in figures 9470-22, 9470-23, and 9470-24. The point of intersection of the pump head capacity curve and the system curve determines the capacity and pressure at which the pump will operate.

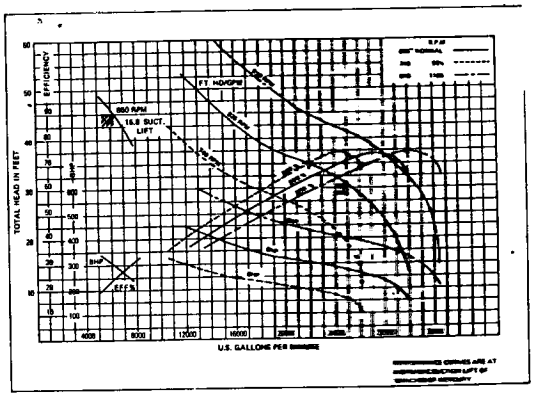


Figure 9470-22

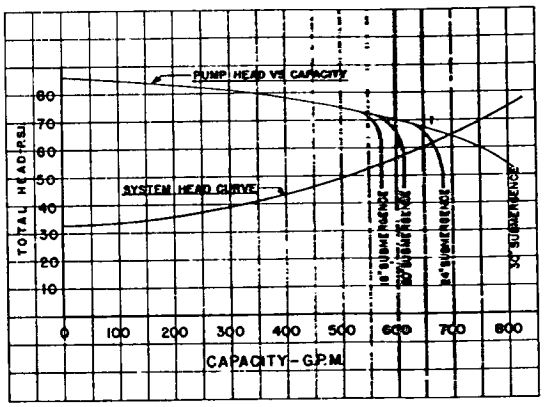


Figure 9470-23

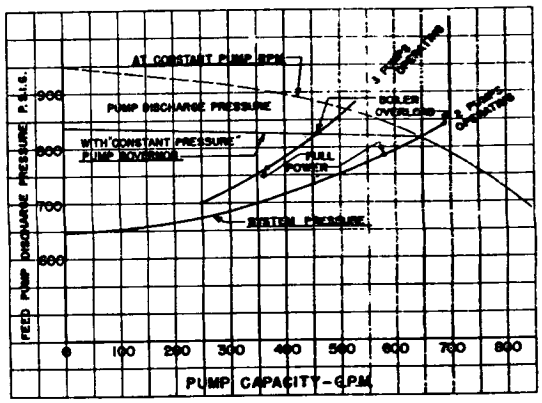


Figure 9470-24

9470.107 SUCTION LIFT

1. Suction lift exists when the total suction head is below atmospheric pressure. It is equal to the static lift plus the losses in the suction piping. When the liquid supply is above the pump center line, suction lift may result if the friction losses are greater than the static head.

2. The ability of a given pump to operate at suction lift depends on such factors as the viscosity, temperature and vapor pressure of the liquid and various design features of the pump itself including the suction passages, the impeller, and its speed. When a high suction lift is required, various means are available to the pump designer to obtain such performance. Liquid velocity in the impeller eye is a major factor but not an exclusive criteria. Pumps for high suction lift are necessarily larger and operate at lower speeds than pumps which can be given ample suction heads.

3. For any given pump the capacity and efficiency are not affected until the suction lift exceeds a certain value. When this value is exceeded, the pump capacity and efficiency drop, the pump will cavitate, and in severe cases may lose prime and seize. Figure 9470-25 shows how suction lift affects performance. This curve is for a fire and bilge pump.

9470.108 CAVITATION

Cavitation in a centrifugal pump occurs when the absolute pressure in the liquid at any point in the pump drops to or below its vapor pressure. When such a condition exists, vapor bubbles form and later collapse as soon as subjected to higher pressure at some other point in the flow. Cavitation in a pump may result from improper pump design, high suction lift, improper pump selection for the specific suction conditions to be encountered, and improper suction piping arrangement. Cavitation results in noise and vibration of the pump, reduction in maximum capacity and efficiency, and in pitting and corrosion of the pump parts, especially the impeller. Where cavitation cannot be avoided, it is necessary to provide special materials for the impellers and other parts to resist cavitation pitting. Naval condensate pumps, for example, operate nearly all of the time in cavitation and the relationship between capacity and submergence makes them selfregulating as to capacity. (See article 9470.149 and figure 9470-26.)

9470.109 NET POSITIVE SUCTION HEAD

"Net positive suction head" or NPSH is the suction pressure measured from absolute zero minus the vapor pressure expressed in feet of liquid under the actual pumping conditions. It represents the amount of energy available at the

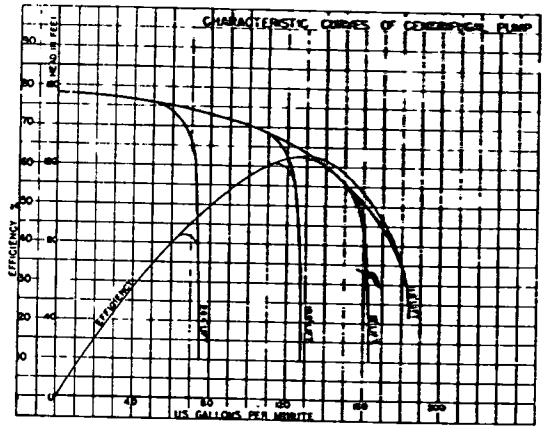


Figure 9470-25

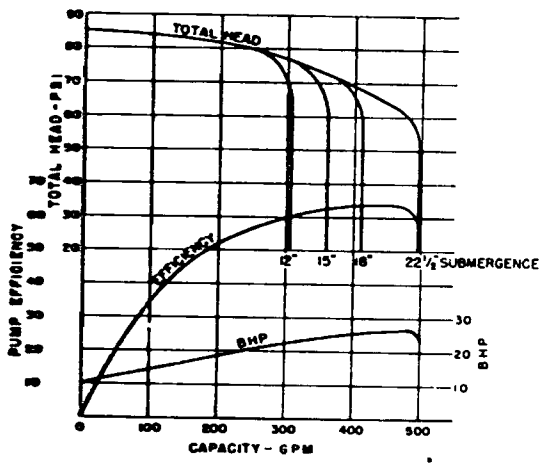


Figure 9470-26

pump suction to get liquid to flow into the pump. Two types of NPSH must be considered in selecting and installing pumps:

1. Available NPSH represents that of the system and is based on knowledge of the suction piping, location of pump with respect to the supply, pressure, temperature, etc.

2. Required NPSH is a condition of the pump design, usually as based on actual tests of nearly identical pumps, and indicates the minimum suction head minus the vapor pressure which is necessary to overcome inlet losses within the pump and ensure rated performance under satisfactory operating conditions. Both available and required NPSH vary with pump capacity.

9470.110 SERIES OPERATION

Centrifugal pumps may be arranged to operate in series when the required pressure exceeds that of one available pump or where it is necessary for a booster pump to be installed between the source of supply and the service pump. Examples of such installations on naval ships include fresh water booster pumps on larger ships, diesel engine circulating water pumps in which a booster pump is located below the water line to supply the engine attached pump on a higher deck, and the feed booster pumps installed between deaerating feed heaters and feed pumps. The main points to watch out for in such installations are:

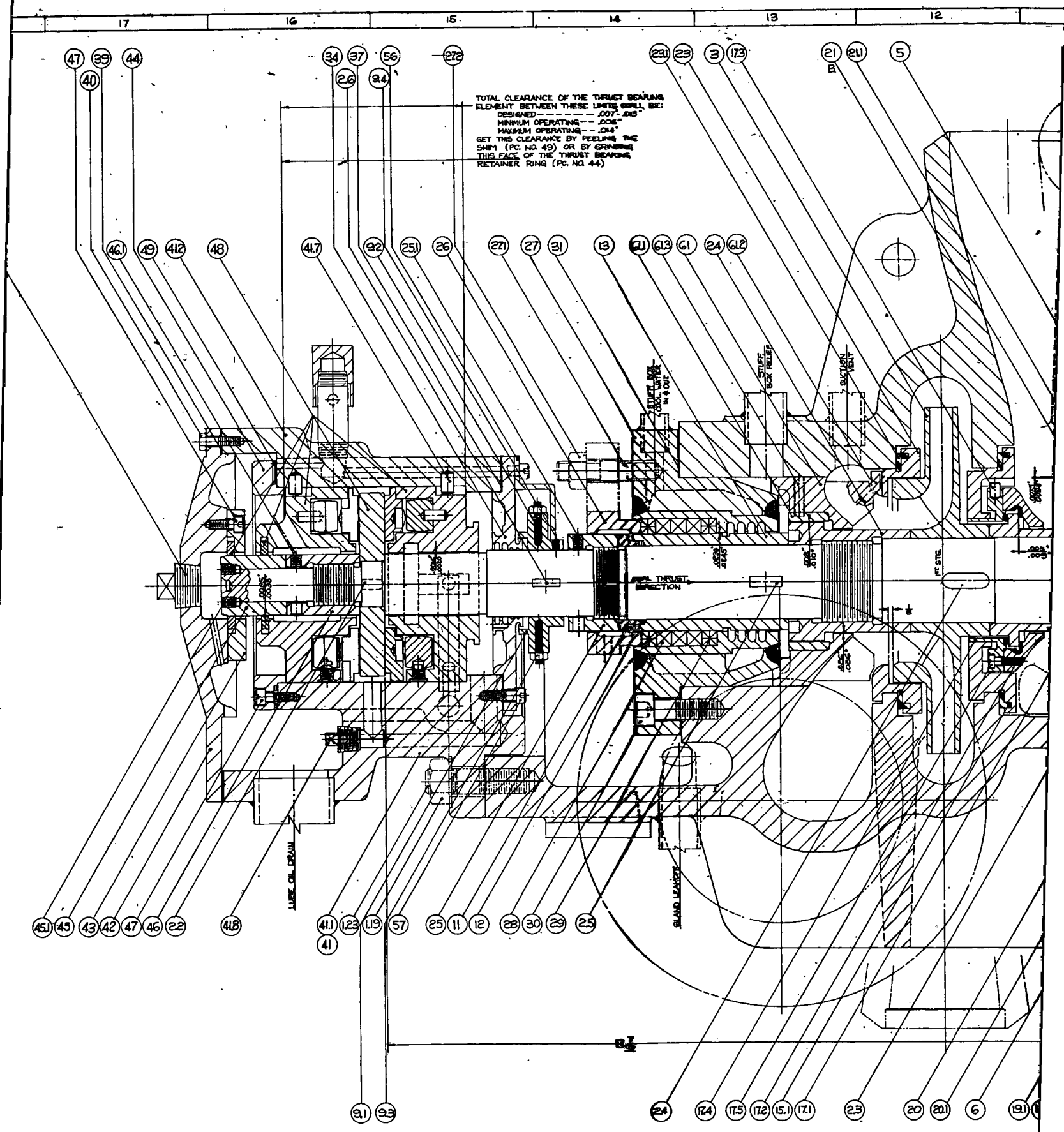
1. The booster pump (acting as the first stage) should have identical or slightly greater capacity rating.

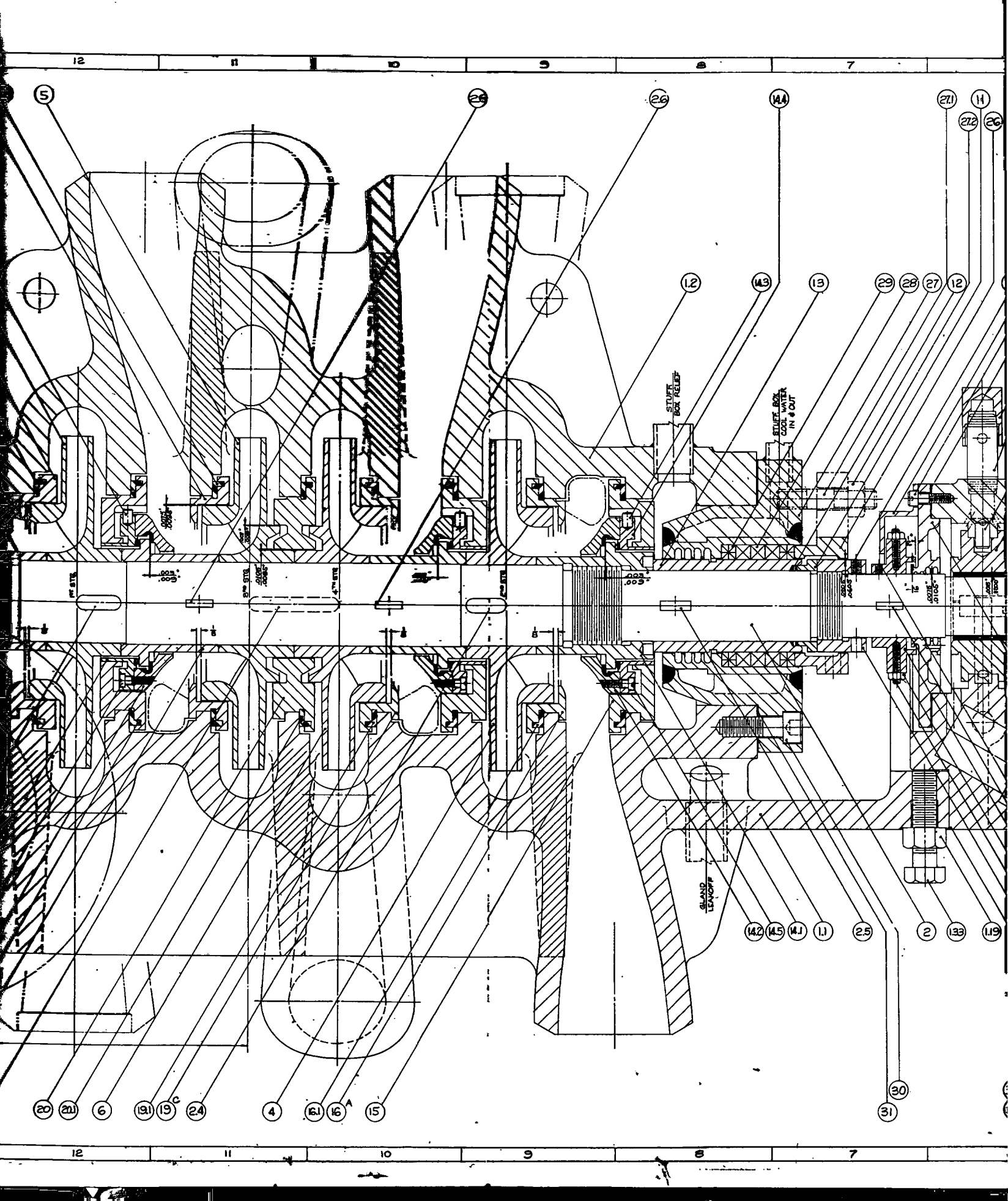
2. The maximum capacity of the first pump should equal or exceed the second.

3. The booster pump should always be started first and secured last.

9470.111 PARALLEL OPERATION

Pumps may be installed to operate in parallel when it is desired to have several smaller units to handle more efficiently varying capacity demands over a wide range, or when maximum reliability is required. Best examples are main and auxiliary condensate pumps, feed booster pumps, and boiler feed pumps. For satisfactory and stable operation in parallel, pumps should have constantly rising head from rated capacity to shutoff. Where pumps of different capacity ratings are to operate in parallel, as for example


TOTAL CLEARANCE OF THE THRUST BEARING ELEMENT BETWEEN THESE LIMITS SHALL BE:
DESIGNED - - - - - - - .007" - .015"
MINIMUM OPERATING - - - .006"
MAXIMUM OPERATING - - - .014"
GET THIS CLEARANCE BY PEELING THE SHIM (PC. NO. 49) OR BY GRINDING THIS FACE OF THE THRUST BEARING RETAINER RING (PC. NO. 44)
STUFF. BOX COOL. WATER IN & OUT
STUFF. BOX RELIEF
SUCTION VENT
THRUST BEARING
LUBE OIL DRAIN
STAND LEAKOFF
47 39 44 40 34 37 56 27.2 23.1 23 3 17.3 21 21.1 5
46.1 49 41.2 48 41.7 9.2 25.1 26 27.1 27 31 13 61.1 61.3 61 24 61.2
45.1 45 43 42 47 46 22 41.8 41.1 12.3 11.9 57 25 11 12 28 30 29 25
41
9.1 9.3
24 17.4 17.5 17.2 15.1 17.1 23 20 20.1 6 19.1


12
11
10
9
8
7
STUFF. BOX RELIEF
STUFF. BOX COOL WATER IN & OUT
GLAND LEAKOFF
1ST STG.
3RD STG.
4TH STG.
2ND STG.
5
26
26
14.4
27.1
11
27.2
26
1.2
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29
28
27
12
14.2
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1.1
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13.3
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19
C
2.4
4
16.1
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A
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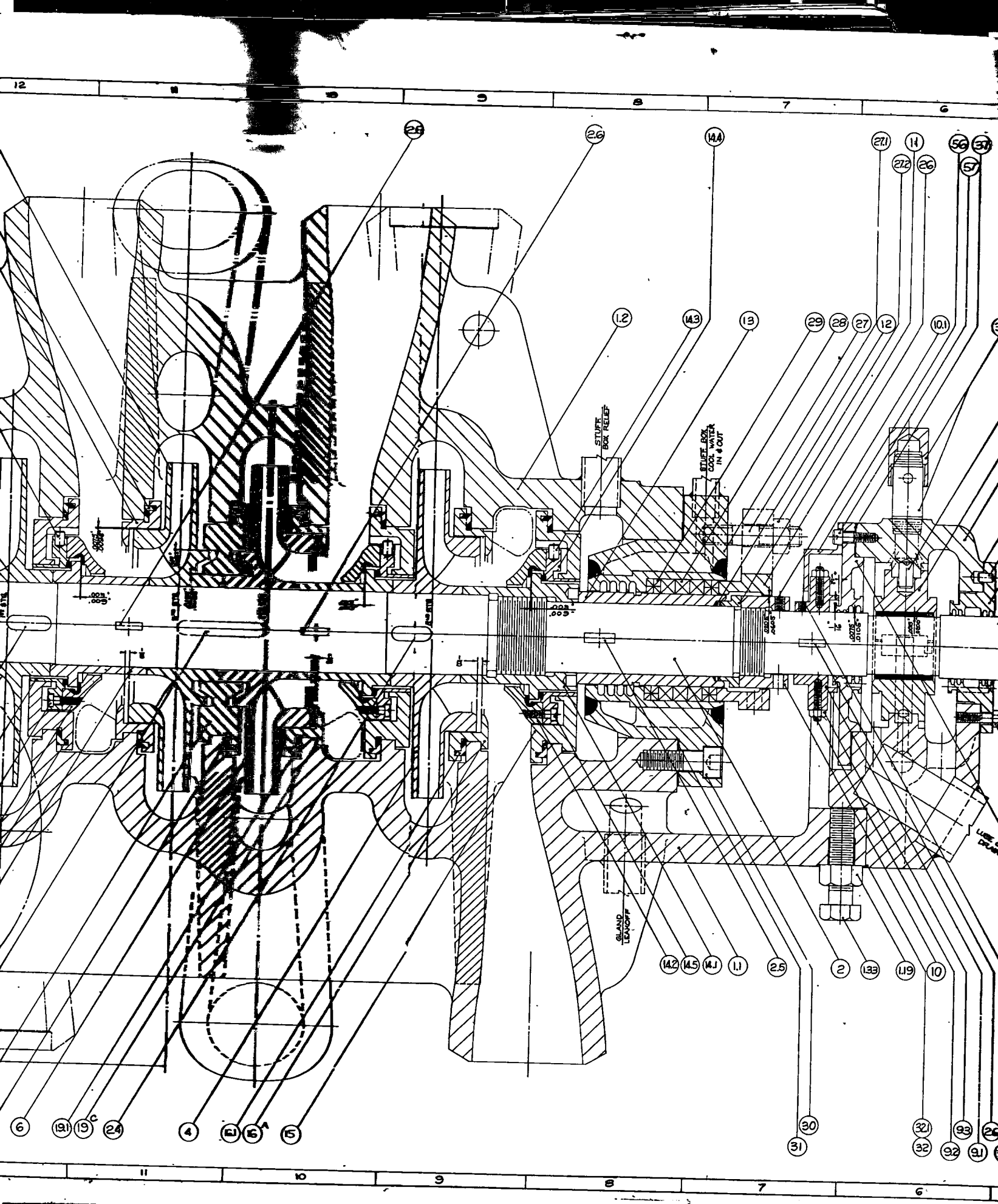
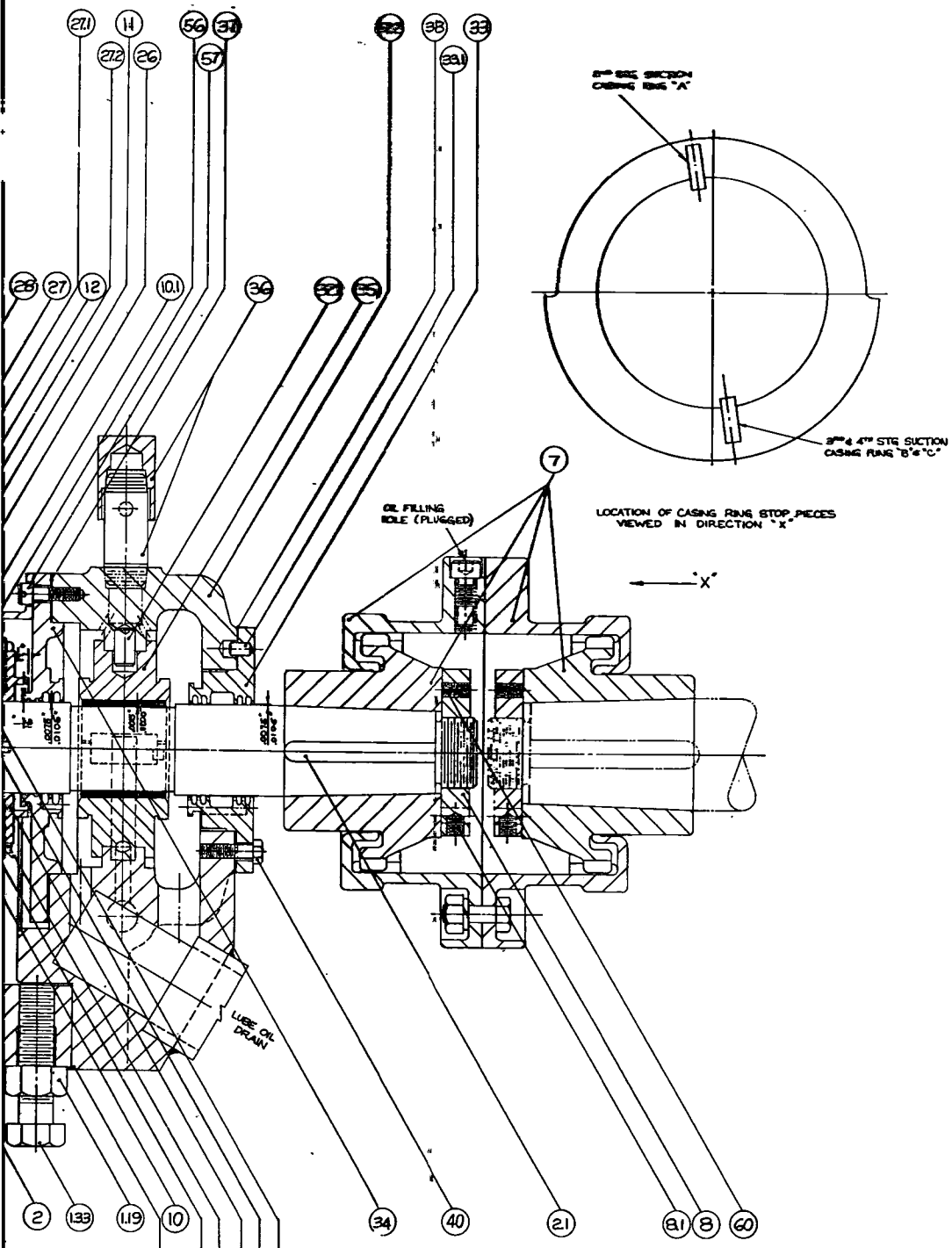
31


STUFF BOX RELIEF
STUFF BOX COOL WATER IN & OUT
GLAND LEAKOFF



GENERAL NOTES

NOTE 1: AT ASS'Y, PEEN OVER ALL C'BORES FOR SOCKET HEAD CAP SCREWS, & STAKE ALL C'BORES AT THE SLOTS OF THE FILISTER HEAD MACHINE SCREWS

REFERENCES

ITEM NO.	TITLE	
1	"HBL" TYPE FOUR STAGE CENTRIFUGAL PUMP: SECTION & LIST OF DETAILS & SUB-ASSEMBLY REF. PLANS	
2	LIST OF MATERIAL	
3	THRUST BRG. HOUSING ASS'Y	
4	LINE BRG. HOUSING ASS'Y	
5	CASING ASS'Y	
6	STUFFING BOX ASS'Y	
7	SHAFT ASS'Y	
8	CASING RING, 1ST STAGE	
9	THRUST BRG. HOUSING COVER	
10	LINE BRG. BUSHING	
11	INTERSTAGE RING	
12	COUPLING, COMPLETE	
13	GLAND, ASS'Y	
14	LABYRINTH SLEEVE, OUTBOARD & INTERM.	
15	SHAFT SLEEVE	
16	PRESSURE BREAKDOWN SLEEVE	
17	LABYRINTH SLEEVE, INBOARD	
18	CASING RING, 2ND, 3RD & 4TH STG.	
19	LABYRINTH RING	
20	THRUST BRG. RETAINER RING	
21	LABYRINTH BUSHING	
24	LABYRINTH HOUSING	
25	LINE BRG. COVER	
26	BEARING COVER	
27	SLIP-ON FLANGE (B-110)	
28	SEAL RING	
29	IMPELLER, 1ST & 2ND STG.	
30	IMPELLER, 3RD & 4TH STG.	
31	150# SOCKET TYPE WELD FLANGE-FEMALE	
32	150# SOCKET TYPE WELD FLANGE	
33	THRUST COLLAR NUT	
34	SEAL RING RETAINER	
35	SHAFT SLEEVE NUT, R.H. INBOARD	
36	SHAFT SLEEVE NUT, L.H. OUTBOARD	
37	SHAFT SLEEVE GLAND	
38	COUPLING NUT	
39	SEAL RING SLEEVE	
40	VENT, COMPLETE	
41	WASHER	
42	LAMINATED SHIM	
43	4 1/2" (4 & 4) COMBINATION DOUBLE THRUST & RADIAL BEARING, COMPLETE	
44	BALANCING RING, ASS'Y	
45	BALANCING RING COVER	

Figure 9470-27

main and auxiliary condensate pumps, the shutoff head of the smaller pump should be slightly in excess of the shutoff head of the larger pump. This is essential in order that the smaller units are not blocked off the line by the higher pressure developed by one of the larger pumps with which it may be paralleling.

9470.112 BOILER FEED PUMPS

1. Boiler feed pumps, in general, are of the high speed, multi-stage type, driven by steam turbines. Most pumps are horizontal; however, vertical steam turbine and motor driven pumps are used where space is limited. Figure 9470-27 shows a typical destroyer feed pump.

2. Chapter 9560 describes the different types of feed systems in general use. The pressure closed system, figure 9560-10, is installed on most ships of the active fleet. Feed pumps for this system handle water at 225° to 250° F. temperature and with suction pressures of approximately 50 psi gage. This is equivalent to an available NPSH of 110 to 85 feet respectively. The required NPSH is in the order of 20 feet. Suction pressure is usually supplied by a feed booster pump since the feed pumps operating at high speed would not have sufficient available NPSH. In some auxiliary ships the feed heater is located at a sufficient height above the feed pump to ensure adequate suction head without a feed booster pump. Boiler feed pumps require an NPSH of sufficiently high margin to take care of momentary or sudden increase of pump capacity.

3. Feed pumps require recirculation from discharge to source of supply, preferably the feed tank or deaerating heater, to protect the pump from flashing when operating at low capacity. See figure 9470-28 for effect of low capacity on temperature rise within the pump.

4. Modern feed systems involve the pumping of extremely pure unbuffered feed water of low pH values. In order to withstand the corrosive action of such feed water, boiler feed pumps are now constructed of corrosion resisting chrome or chrome nickel alloy steel.

9470.113 FEED BOOSTER PUMPS

1. Feed booster pumps are generally vertical, single or two-stage pumps driven by steam turbine or electric motor. Their construction is very similiar to condensate pumps. Feed booster pumps have low suction and impeller eye velocities and operate at low rpm because of the low NPSH available. Such pumps take suction from the deaerating feed tank under a submergence of 6 to 12 feet and this static head is practically the equivalent NPSH.

2. Feed booster pumps should be fitted with suction vent connections and with a recirculation line from discharge back to the feed heater. This line should be left open when starting or securing and when pump is discharging to a reciprocating emergency feed or main feed pump. At other times the recirculation from the centrifugal main feed pump will give the booster pump sufficient protection against overheating at low capacities.

9470.114 CONDENSATE PUMPS

1. Main condensate pumps are vertical, one or two-stage pumps driven by steam turbine or electric motor. Condensate pumps take suction under extremely low submergence from the hot well of the condenser. First-stage impellers are usually single top suction type and the suction nozzle and piping are arranged to have a continuous downward slope from condenser to impeller eye. Every precaution is taken during the machinery design to ensure that the impeller is submerged even at maximum angle of ship's roll. Otherwise a severe water hammer may occur which has resulted in distortion of impeller shrouding between vanes. First stage impeller eye is vented back to the vapor space of the condenser.

2. Sealing water for condensate pump stuffing boxes should be taken from a remote source to ensure a continuous supply of clean water under adequate pressure to prevent leakage of air into stuffing box. Condensate pumps should not be operated at low capacities for very long without recirculation since their low margin of NPSH available over that required is insufficient to withstand temperature rise of more than a degree or two without flashing.

3. A typical two-stage condensate pump is shown in figure 9470-29.

9470.115 MAIN CONDENSER CIRCULATING PUMPS

Pumps for main circulating service on high-speed combatant ships are generally of the vertical propeller type driven by direct connected or geared steam turbine and are of capacities of 10,000–40,000 gallons per minute at total heads of approximately 30 feet. For such installations circulating pumps are usually arranged to operate when maneuvering, backing, or going ahead at low speed. At high speeds scoop circulation is adequate. For auxiliary ships, main circulating pumps are usually vertical, centrifugal, double-suction type driven by electric motor. Pump size is usually under 10,000 gpm.

9470.116 FIRE PUMPS

Fire pumps are usually horizontal, singlestage, double-suction type, either electric motor or steam turbine driven. Emergency fire pumps are usually diesel engine driven. Fire pumps are installed in numbers, locations, and sizes as necessary for best ship protection. Capacities may cover a range of 100–250 gpm for small ships, and up to 2,000 gpm for large ships.

9470.117 MISCELLANEOUS CIRCULATING AND SUPPLY PUMPS

1. The majority of pumps for such applications as flushing, auxiliary circulating, refrigerator circulating, distiller circulating, fresh water, etc., have capacities up to about 2,500 gpm at total heads under 100 psi. The horsepower seldom exceeds 25. Such pumps are usually of the single-inlet, centrifugal, volute type, close-coupled to the driving unit, which is usually an electric motor. In close-coupled units the shaft of the motor extends into the pump casing, with the impeller secured to the extended motor shaft; the pump casing is bolted to the end bell or bearing bracket of the motor (figure 9470-30). The pump casing is fitted with a packing box in way of shaft and a cored type packing gland with a drip connection is usually provided. In addition the shaft is always fitted with a water flinger between the packing gland and the motor bearing bracket, the object of which is to prevent the water leakage from the packing box from following the shaft and entering the motor bearing and the motor housing. Frequently the pump casing is provided with supporting feet in addition to being bolted to the motor

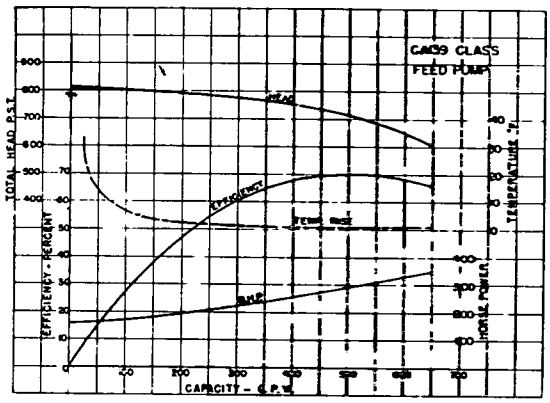


Figure 9470-28

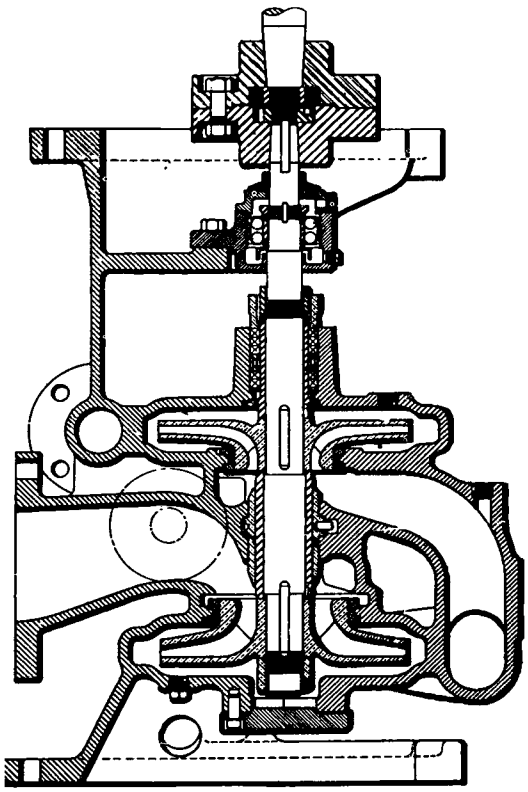


Figure 9470-29

although this construction is not essential where the shaft overhang beyond the motor is short.

2. In 1945, the Bureau of Ships introduced a line of Navy standard close-coupled pumps. At present this line includes 12 different size casings with motors to suit. The standard pumps are interchangeable as to size and location of suction and discharge connections and foundation bolting. Further motors are interchangeable as to shaft extension and bolting to pump casing so that motors of different manufacture are interchangeable on pump casings of the same size. The Navy standard close-coupled pumps will eventually be stocked in adequate quantities to permit their use as replacement for worn out nonstandard pumps. However, no large scale replacement program will be authorized except in instances where a specific unsatisfactory installation warrants general replacement.

9470.118 SHIP'S FRESH WATER PUMPS

Fresh water pumps are generally of the close-coupled type. They have a special feature of being automatically controlled by a pressure switch to maintain pressure in the fresh water system within predetermined limits. For this reason fresh water pumps should have a rather steep head-capacity curve so that pump is still delivering an appreciable flow as it builds up to the stop pressure.

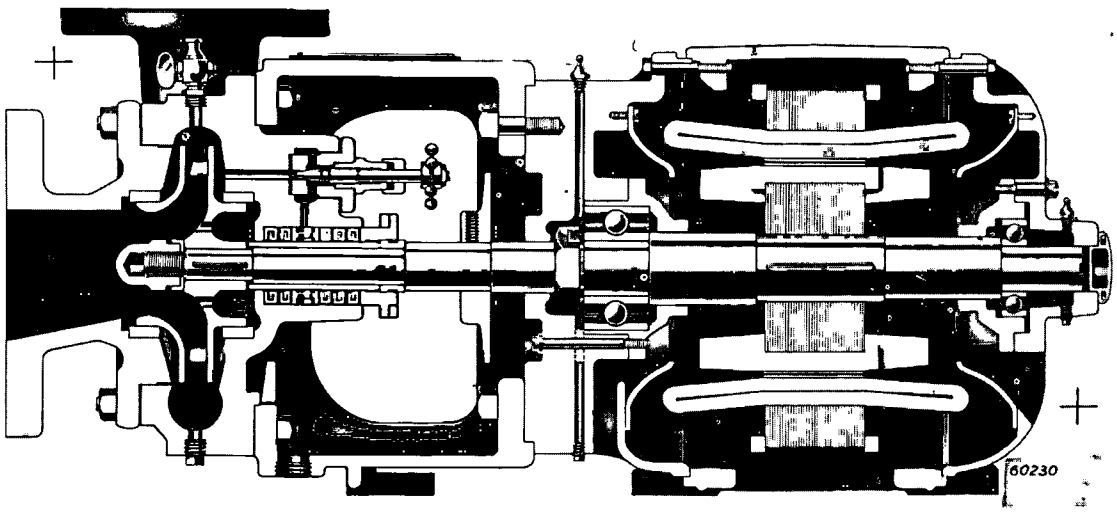


Figure 9470-30

9470.119 DISTILLING PLANT PUMPS

1. The following basic pump services are required for low pressure distilling plants:

a. Distiller circulating pump.
b. Distiller condensate pump.
c. Brine overboard pump.
d. Evaporator coil drain pump.

Additional services required for certain sizes and plant installations are:

e. Evaporator feed pumps.
f. Distiller fresh water pumps.

2. Pumps a., e., and f. above are usually horizontal close-coupled type. Pumps b., c., and d. are condensate type pumps and may be any one of the horizontal close-coupled, vertical-coupled, or vertical close-coupled types. Recent installations favor use of vertical pumps for distiller condensate, brine overboard, and coil drain service to ensure adequate suction submergence and to keep the motors out of unsatisfactory locations where they are subject to splash and drip.

3. Distiller condensate, brine overboard, and coil drain pumps should be either selfventing by installation of a suction pipe which rises continuously without pockets from the pump suction to the evaporator shell or condenser. Where this is impractical, a separate suction vent connection should be provided, led to a point on the condenser or evaporator shell well above the highest water line.

9470.120 LUBRICATING OIL AND CARGO OIL PUMPS

1. Lubricating oil or cargo oil pumps involve the pumping of viscous liquids which have the following effect on the performance of a centrifugal pump.

a. Head and capacity are reduced over that obtained when pumping water except that shutoff head is practically the same.

b. Efficiency is lowered and the horsepower for a given capacity is greater.

2. These changes in performance are primarily due to the increased disk friction losses of the impeller. Figure 9470-31 shows the effect of pumping oils of various viscosities on a main cargo pump. Nevertheless, centrifugal oil pumps have advantages in weight and space for the large pumping capacities now provided in modern tankers. Main lubricating oil pumps of the centrifugal type have been installed in several recent ships. Their viscosity range is usually between 130 and 500 SSU; therefore, the performance difference compared to water is not so marked. However, such pumps are required to handle large percentages of entrained air and special features are required to prevent difficulty due to air separation in the impeller eye or vane inlets.

9470.121 REGENERATIVE PUMPS

Regenerative pumps, also known as "peripheral" or "turbine" type, are occasionally used in naval service for applications involving low capacity and relatively high pressure. They are particularly useful on fresh water system in small ships since they have a steep head curve. The development of pressure does not follow the laws set forth in article 9470.105 and the flow through the pump is of a spiral path between the vanes on the impeller periphery and the

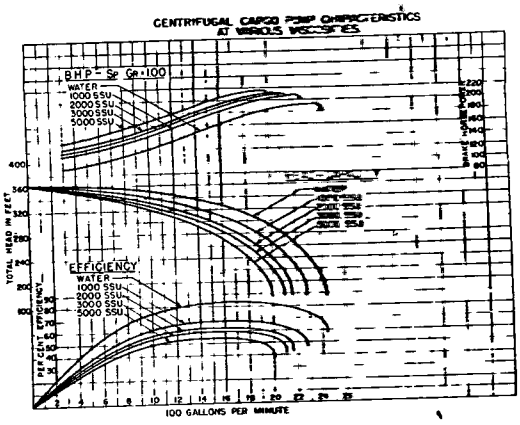


Figure 9470-31

casing raceway. Due to the close clearances only clean water services are recommended and their clearances must be carefully maintained. This type of pump is able to handle large percentage of vapors. Typical construction of a regenerative pump is shown in figure 9470-32

9470.122 PORTABLE SUBMERSIBLE PUMPS

1. Motor-driven, portable, submersible pumps of the centrifugal type are provided for all ships of the fleet and fleet auxiliaries and are intended primarily for damage control purposes, Some pumps of this type are stowed semi-permanently, with suction and discharge hose connected, for use as drainage pumps. The number of these pumps allowed per ship is covered in each ship's allowance list.

2. Portable submersible pumps are furnished with either direct current or three phase alternating current motors as required. The pumps have the following minimum performance:

Capacity (gpm)	Total Head (feet)
140	70
180	50

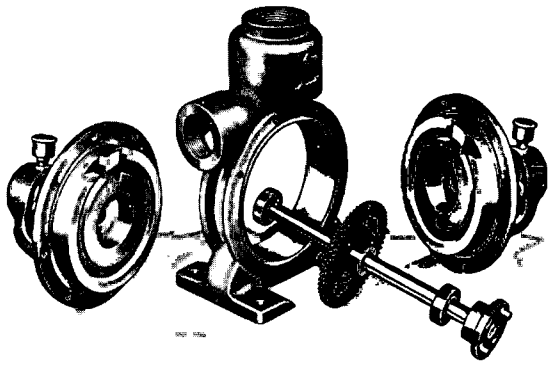


Figure 9470-32

The total heads include 20 feet suction lift to which must be added hose friction and vertical discharge lift to determine the suitability of the pump for the required operation. Where the total head required exceeds the head available in the pump, two pumps may be coupled for series operation.

3. Each pump is furnished with 75 feet of flexible cable having the controller installed in the cable 30 feet from the pump. Suction and discharge connections are arranged for Navy standard 2-1/2-inch fire hose. All pumps are designed to permit passage through a hole 10 inches in diameter. Each pump is provided with a suction strainer which should always be attached when the pump is in use. Each pump is also provided with a foot valve for attachment to the suction hose when the pump is to be operated above the water pumped. Priming is required when the pump is operated unsubmerged and this may be accomplished by lowering the pump into the water to start it, or by filling the pump casing and suction hose through the discharge hose connection. An eye is provided on the discharge end of the pump for attachment of a line for handling the pump.

9470.123 PORTABLE ENGINE-DRIVEN PUMPS

Gasoline engine-driven portable pumps classified as P-250: (250 gpm at 100 psi) are furnished to ships of the fleet and fleet auxiliaries. These pumps are dual service pumps which can be used as either a fire or drainage pump. As a drainage pump, they may be used in dewatering flooded spaces. Operation, care, and maintenance of these pumps are fully described in chapter 9930, Naval Ships Technical Manual.

Part 2. Operation

9470.131 USE OF MANUFACTURERS' INSTRUCTION BOOKS

The instructions contained in this chapter for the operation, maintenance, and repair of pumps are general. All pumps cannot be covered because of the great number of diverse makes and types and designs encountered in the naval service. For all but the simplest types of pumps, manufacturers' instruction books are furnished with the plans. These instruction books contain detailed information concerning the operation, maintenance, and repair of the specific pump installed and should be studied carefully before attempting to operate or service the unit. If any conflict exists between the instructions given in this chapter and the manufacturers' instructions, the Naval Ships Systems Command should be consulted.

9470.132 PREPARING PUMP FOR OPERATION

Prior to starting a centrifugal pump for the first time or after an overhaul, proceed as follows:

1. Check alinement at coupling and if necessary bring back into correct alinement. (See articles 9470.173 and 9470.185.) Before reassembling check direction of motor rotation to be sure it is correct as indicated by arrow affixed to pump casing.

2. Clean all external surfaces of the pump and driving unit to remove all dirt that may have accumulated during installation.

3. Check all auxiliary piping including lubricating lines, water cooling lines, vent lines, bypass piping, gage connections, priming connections, etc.

4. Examine all bearings to be sure they are properly lubricated.

5. Make sure stuffing boxes are properly packed.

6. Stuffing box glands should not be drawn too tight. A slight flow of water through the stuffing box is necessary for proper lubrication of the packing.

9470.133 STARTING

To start a centrifugal or axial flow (propeller) type pump, proceed as follows; subject to more detailed instructions which appear later in this section:

1. Check lubricating-oil level in pump tank or bearing housings. Fill oil cups or reservoirs, if fitted. If lubricated by a detached pump, open and adjust all oil delivery and return valves.

2. Open steam and exhaust casing drains of turbine or driving engine.

3. Open valves on pump to gland water seals, where fitted. Slack up on packing in pump-stuffing boxes to permit a reasonable amount of leakage.

4. Open steam and exhaust root valves.

5. If pump is driven by a reciprocating engine, jack it into position for starting. Remove starting bar.

6. Open turbine or engine exhaust valve.

7. Lift all relief and sentinel valves by hand.

8. Open pump suction valve.

9. Prime the pump. (See article 9470.141.)

10. For main feed pumps, and other pumps where fitted open recirculating (bypass) line valves and gland leak-off line valve.

11. Start unit.

a. Crack turbine throttle valve sufficiently to free lines, steam chest, and exhaust casing of water, being careful that turbine runs, if at all, at very slow speed. As soon as turbine is free of water, close drain, then open throttle fast enough to bring unit up to speed gradually.

b. For reciprocating engines, crack the throttle to allow cylinder to warm up slowly; then open throttle slowly. Sometimes it may be necessary to open the throttle smartly in order to start the engine, but if this is done, care must be taken to choke down as soon as engine starts, and to run slowly until engine is warmed up. Close drains when engine is clear of water.

c. For motor-driven pump, start motor. (See instructions for operation of controllers and motors, chapter 9063.)

12. Open vent valves or air cocks on pump casing to check that all entrained air has escaped.

13. After all air has escaped, close vent valves or air cocks on pump casing.

14. For condensate pumps or other pumps vented back to a vacuum, open the valve in the vent connection.

15. When pump is up to proper speed, open the discharge valve. For axial flow (propeller) type pump, it is necessary to start with the discharge valve open to avoid either:

a. Driving unit frequently too small to start pump against closed discharge, or

b. Excessive discharge head may be developed.

16. Check lubrication system to see that all bearings are getting a proper supply of oil and that oil pressure and temperature are correct.

17. Check all gages to see that proper pressures are being developed.

18. For turbine-driven units, check revolutions per minute to see that rated speed is not exceeded by more than five per cent.

19. Set up on packing glands slowly until leakage is reduced to a mere trickle.

9470.134 STOPPING AND SECURING

To stop and secure, proceed as follows:

1. Close pump discharge valve.
2. Close steam throttle valve or stop motor.
3. Close exhaust valve.
4. Close vent valves.
5. For main feed pumps and others, where fitted, close recirculating (bypass) and gland leak off line valves.
6. Close valves on pump to gland water seals.
7. Close pump suction valve.
8. Open turbine or engine drains.
9. Close all delivery and return oil valves when pump is lubricated by a detached pump.
10. Close steam and exhaust root valves.
11. Close steam drains after turbine or engine is completely drained.

9470.135 CENTRIFUGAL PUMP TROUBLES

The following points should be checked in the event the pump does not operate normally or satisfactorily:

1. No liquid delivered.
   a. Pump improperly primed.
   b. Pump speed too low.
   c. Discharge head too high. Valves may be partly closed or other obstructions impose too high a system head.
   d. Suction lift too high. Unless pumps are specifically designed for high suction lift, the normal design is limited to approximately 15 feet of water suction lift when handling cold water.
   e. Impeller passages plugged up.
   f. Pump running opposite to correct rotation.
2. Not enough liquid delivered.
   a. Air leaks in suction pipe.
   b. Air leaks in stuffing boxes.
   c. Pump speed too low.
   d. Suction lift too high.
   e. Suction sea chest or connection to supply not submerged enough.
   f. Impeller passage partly clogged.
   g. Discharge head too high.
   h. Not enough suction pressure when pump handles hot or volatile liquids. (Indicated by noise and fluctuating pressure.)
   i. Mechanical defects including wearing rings worn, impeller damaged or eroded, stuffing box packing and/or sleeves worn and need replacing.
3. Not enough pressure.
   a. Pump speed too low.
   b. Air or gas in liquid pumped.
   c. Mechanical defects as noted in 9470.135.2 above.
4. Pump works for a while and then fails to deliver.
   a. Air leakage in suction line.
   b. Air leakage in stuffing box.
   c. Stuffing box water seal plugged.
   d. Suction supply uncovered.
   e. Suction lift too high.
5. Pump takes too much power.
   a. Head lower than pump rating. This results in pump handling a capacity greater than rated and it will require more power (figure 9470-21).
   b. Pump speed too high.
   c. Specific gravity or viscosity of liquids other than water may be higher than pump designed to handle.
   d. Mechanical defects including rotor is binding, shaft bent, stuffing boxes too tight, wearing rings worn, misalinement.

9470.136 LUBRICATION

1. Lack of lubrication is one of the primary causes of pump failures. Pump bearings are lubricated by one of the following methods:
   a. Ring oiled sleeve bearings.
   b. Grease lubricated ball bearings.
   c. Oil lubricated ball bearings.
   d. Forced feed lubrication.
2. Check ring oiled bearings to see that oil rings are rotating freely and are carrying an adequate amount of oil. Check oil level frequently during each watch.
3. Reciprocating engine-driven pumps are usually lubricated by either sight feed drip cups or wick lubricators. See that oil cups are filled with oil and that adequate oil supply is being fed to bearings; check frequently during the watch.
4. Motor-driven pumps and some turbine-driven pumps fitted with ball bearings are usually fitted for grease lubrication. Before starting, see that all grease cups and bearing housings are filled with the proper amount of lubricant and that no water or foreign matter is in the bearing housing. Grease lubrication is used for the two fold purpose of lubricating the bearing and excluding water and foreign matter from the bearing housing. Pump shafts are usually fitted with a water flinger between pump shaft stuffing box gland and bearing housing. See that such flingers are effectively preventing water from the pump glands from following along the shaft and entering the bearing housing. Occasionally it will be found that sleeves fitted on pump shafts do not fit the shaft tightly and water can leak under the shaft sleeves. If such leakage exists, care should be taken to prevent water from entering the bearing housing.
5. Turbine-driven pumps are usually fitted with a self-contained force lubricating-oil system supplied by an attached lubricating-oil pump of the gear or screw type.
6. Check the oil pressure and oil flow to or from all bearings, including the thrust bearing, being sure that the attached lubricating-oil pump has primed itself. See that cooling water is flowing through the oil cooler or cooling coils in the oil reservoir and that all air is vented from the water side of the oil cooler. It may be necessary to free the lubricating-oil system of air in order to maintain a steady oil pressure. This can be accomplished by opening air cocks on the high points of the lubricating-oil system. These should be closed immediately when oil appears. Check oil reservoir to see that it is free of water. Check bearing housings to see that no water from pump or turbine glands is getting into lubricating-oil system.

9470.137 STUFFING BOXES AND PACKING

1. The stuffing boxes on a centrifugal pump serve two purposes. When the pump is working under a positive suction head, the purpose of the packing is to prevent liquid from leaking out of the pump. When pressure at the inner end of the stuffing box is below atmosphere, the stuffing box must prevent air from leaking into the pump.

2. When a pump is designed to work on a suction lift, stuffing boxes are fitted with lantern rings or seal cages usually located near the center of the stuffing box to prevent air leakage and to lubricate the packing. Sealing liquid is injected to this seal cage.

3. When the pump handles clean, cold water, the seal is usually connected to volute casing of the pump. On multistage pumps the connection is to the first stage to avoid excessive pressure.

4. An independent external liquid sealing supply is desirable under any of the following conditions:

a. High suction lifts.
b. Low discharge pressure.
c. Condensate pumps.
d. Hot water.
e. Liquid contains abrasives or foreign matter.
f. Corrosive or volatile liquids are handled.

5. Particular care must be taken to ensure that water piping to seals or stuffing boxes is open and clear; otherwise there is a probability that the seal or the stuffing-box packing will wear very rapidly, the shaft will be scored, and the air will be drawn in. Therefore, when the pump is running, a little water should be allowed to trickle out of the stuffing box. The efficiency of the seal may be tested with a candle flame to detect if air is being drawn into the pump. This should not be tried on a pump handling oil, gasoline, or other inflammable liquid.

6. Even with pump under a positive suction head, water should be allowed to drip from the stuffing box in order to lubricate and cool the packing.

7. Packing should be occasionally changed.

9470.138 VENTS

The petcocks at the highest point of the discharge and at the pump casing may need to be opened frequently to release entrained air. If so, it is a good plan to keep them cracked so as to allow a continuous leakage of air and water. This may be piped to a proper drain system or to bilge.

9470.139 BYPASS

1. In any centrifugal pump the difference between the shaft horsepower input and the liquid horsepower output is lost in the unit and converted to heat. This frequently results in unacceptable temperature rises at low capacities particularly for high pressure pumps. Boiler feed pumps, and in some instances fire pumps, which may develop high temperatures due to operating at low capacity or shut off for substantial periods are protected by a bypass from pump discharge back to the source of suction supply. In the case of feed pumps, this bypass is led from pump discharges through an orifice back to the deaerating feed tank or feed heater. The orifice is designed to bypass the minimum quantity of water necessary to protect the pump and may vary from 5 to 15 percent of pump rated capacity depending on the individual pump characteristics. Fire pumps are generally protected by a bypass of one to two percent.

2. The bypass line on feed pumps must be kept open at all times the pump is operating. The bypass line on fire pumps may be manually controlled and opened when pump is to be operated at or near zero capacity for several minutes or more under standby conditions.

3. Some feed pump installations have been fitted with automatic bypass valves designed to open only as the pump output approaches the minimum permissible capacity. These are mostly of the Hammel-Dahl type and are installed on various Maritime commission tankers converted to naval use.

9470.140 PARALLEL OPERATION

1. When two or more centrifugal pumps are installed and arranged for parallel operation, a stop and a check valve are provided in the discharge of each pump in order that an operating pump may not discharge through one not in operation should the stop valve be left open.

2. Pumps intended to operate in parallel should be designed to have similar characteristics curves with practically the same shutoff head.

3. Operation of several pumps in parallel at very low capacities should be avoided due to the possibility of any unit having slightly lower head being pushed off the line resulting in that unit running at shutoff and developing high temperature.

9470.141 PRIMING

Before starting any centrifugal pump, the pump should be fully primed. The casing and the suction piping must be filled with the liquid to be pumped. There are various ways of priming centrifugal pumps, as follows:

1. If the pump is installed so as to have a positive head on the suction, the pump will prime itself when the vent on the top of pump casing is opened. After all entrapped air has escaped, close the vent valve or air cocks.

2. If the pump operated on a suction lift, prime the pump by the means provided, some of which are as follows:

a. If the suction line has a foot valve or check valve and discharge line contains liquid, fill pump casing by opening bypass valve around discharge valve. Open vent valves or air cocks at top of pump case. Close valves or cocks after all entrapped air has been released. If discharge line has no liquid, open valve to independent supply, if provided, or fill casing by pouring liquid into a funnel inserted in vent connections on casing.

b. If pump is fitted with an air ejector, keep discharge valve closed, open valve to ejector from pump casing, and then turn on steam or air to the ejector. Pump may be started as soon as eductor throws liquid continuously. Ejector should be secured after pump has been started.

c. If pump is connected to a central priming system, open valve to the central priming tank putting vacuum on the pump case.

d. If pump is fitted with or connected to a wet-vacuum pump, follow instructions for the specific unit.

3. If a pump does not build up a pressure after priming, it should be stopped and primed again. While priming, the pump should be turned over by hand to allow air entrained in the impeller to escape.

4. If the pump is the "self-priming" type, then the pump is designed to prime itself automatically by some internal hydraulic device, or by recirculation. This type pump is occasionally installed on smaller vessels. Follow the manufacturers' instructions for the specific unit installed.

9470.142 OPERATION AT BEST EFFICIENCY POINT

1. Centrifugal pumps should be operated as much of the time as possible at or near their design rated conditions which is usually the point of best efficiency. Impeller vane angles and the size of pump waterways can be correctly designed for only one point of operation. For any other point of operation these angles and waterways are either too large or too small.

2. **Excess capacity.** Any pump if operated at excess capacity and lower head will surge and vibrate causing bearings and shaft troubles and take excess power.

3. **Reduced capacity.** When operating at reduced capacity and higher head, the incorrect vane angles cause eddies in the impeller and the casing and will result in erosion in the impeller, casing, and wearing rings. The radial thrust on the rotor increases causing higher shaft stresses, increased shaft deflection and accelerated wear on wearing rings.

9470.143 MAIN CIRCULATING PUMP ON BILGES

In the case of main circulating pumps operating on bilge suction, the pump should be started the same as for main condenser circulating service, then the main injection valve should be gradually closed and the bilge suction valve opened. When the pump is operating on a high suction lift, as when pumping bilges, the speed should be reduced to about two-thirds of rated speed and even when so operated, the pump will be noisy. This cannot be avoided, but can be minimized by slowing the pump.

9470.144 STARTING BOILER FEED PUMPS

1. If the feed system is of the closed type, before starting a main feed pump see that the proper feed booster pump has previously been started and is maintaining the required pressure on the feed pump suction main.

2. If the feed system is of the open, semi-closed type or if the feed heater is connected directly to feed pump suction without booster pumps, see that feed suction line valves at the surge tank or heater is wide open and that water flows freely from pump casing vent valves when open. After the feed pump has been started as previously outlined and has been warmed up, the throttle valve should be opened and the unit brought up to speed slowly until the feed discharge pressure is about 10 psi above the pressure in the combined feed discharge line. Then open the feed discharge valve, or if the stop valve is not fitted, the stop feature of the combined stop check valve on the pump discharge. When this valve is open, the pump discharge pressure should drop to the pressure in the combined discharge line. The pump is then cut in on the line.

9470.145 CHECK VALVE OPERATION

In order to check the operation of the feed pump discharge check valve, close the throttle slightly, lowering the speed and pump discharge pressure to ensure that if pump is tripped off the line, the check valve will close and prevent backflow of water, from the feed discharge mains through the pump, The operation of this check valve can be noted by an indicator if fitted or by hearing or feeling it close. A slight water hammer will often accompany the closing of this check valve. The operation of the check valve may also be determined by noting the pressure on the gages, showing that the discharge pressure of the pump is less than that of the combined feed discharge main.

9470.146 PUTTING FEED PUMP ON LINE

After the check valve has been found to operate satisfactorily, open the throttle, speed up as before, and cut in the pump on the discharge line. If the pump is not fitted for operating with a pressure governor, throttle it to the speed necessary to maintain the required feed discharge pressure. If fitted with a pressure governor, such governor should be put in operation by closing the bypass, or pull-open device, and setting the governor to give the required feed discharge pressure. (See section IV.) The throttle then should be opened wide to ensure that the pump pressure governor takes control. Check all pressures, temperatures, speed, and lubrication to see that normal conditions exist throughout. With the pump pressure governor in operation, the unit should require no further attention insofar as discharge pressure is concerned; however, discharge pressure should be checked carefully to see that it is normal. Where no pump pressure govenor is fitted, the unit will require close attention to throttling to maintain the required feed discharge pressure especially if the ship is maneuvering.

9470.147 STARTING FEED BOOSTER PUMPS

Before starting a feed booster pump, see that booster pump vent line valves are open, valves in suction line from deaerating feed tank to pump are open, discharge valve is closed, and recirculating line valve, if fitted, is open; also that shaft stuffing box sealing water connections, if fitted, are open. Then start pump gradually increasing speed and pressure to normal. Open the discharge valve and see that pressure is built up on the feed pump suction main.

9470.148 STARTING CONDENSATE PUMPS

Before starting a condensate pump, see that there is ample water available in the condenser hot well so that the condensate pump will not run dry. If there is no water in the condenser, about 50 to 150 gallons should be run down to the condenser from the surge tank or feed tank. The pump may then be started as follows:

1. See that condensate pump vent line to condenser shell is open.

2. Open condensate pump shaft packing sealing water connections.

3. If fitted, open recirculating line valve in the recirculating piping from the discharge of air ejector condensers to main or auxiliary condensers.

4. If water to air ejector condensers is supplied by the condensate pump, start condensate pump, then start air ejectors and increase vacuum on condensers.

5. Lower the water level in condensers to normal by allowing surplus to be discharged, taking care that there is sufficient recirculation to prevent the pump from running dry.

9470.149 OPERATING CONDENSATE PUMPS

1. Condensate pumps are generally designed to be self-regulating as to capacity. That is, their capacity is a function of the submergence or height of water in the hot well. Figure 9470-33 is a typical main condensate pump characteristic curve with a super-imposed system head curve. It will be noted that for submergence up to 22-1/2 inches there is a definite maximum capacity for a specific submergence and

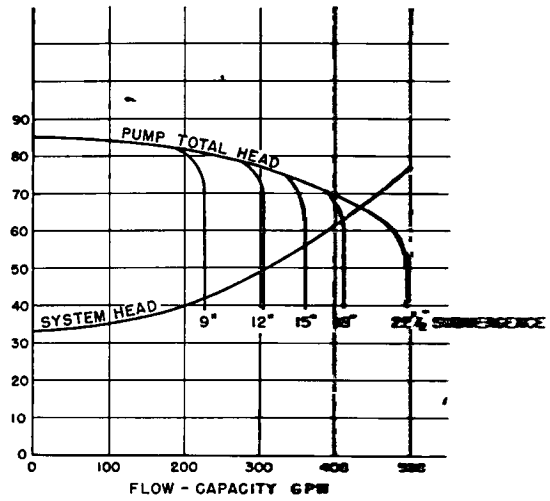


Figure 9470-33

the head-capacity "curve" at this submergence is nearly vertical. If a pump is operated at the maximum capacity for a given submergence and at a lower than normal total head, the pump is cavitating. At least four different basic methods are employed on various naval ships. These methods are:

Method 1. By keeping the discharge valve wide open and varying the pump speed and disregarding submergence entirely.

Method 2. Maintaining constant speed, keeping the discharge valve wide open and disregarding submergence entirely.

Method 3. Maintaining constant speed and throttling the discharge valve in order to keep the liquid level in the condenser visible in the gage glass.

Method 4. Maintaining constant speed, discharge valve wide open, and recirculating to maintain constant submergence.

2. Because of the dissimilarity of condensate pump installations and their related piping systems on various naval ships, the Naval Ships Systems Command will not standardize by hard and fast rules any one method to be used. It is considered that many factors other than operation of the pump alone must be given consideration in establishing operating methods for condensate pumps and systems on each particular ship. However, it is recommended that methods 2 and 4 are preferable for reasons discussed below.

Method 1. (See figure 9470-34.) Considered the worst of all methods of operation. When operating at reduced rpm along head capacity curve H', pump cannot respond to increased load beyond point B on the system curve S with increase in speed. This requires constant manning of the throttle to handle variations in capacity.

Method 2. (See figure 9470-35.) Preferred to all other methods provided pumps are not over-designed or overrated as to head, and provided the combination of pumps and systems can be demonstrated to work satisfactorily without adjustment for variations in load. Pump at reduced capacity will operate in cavitation at the intersection of the system curve S and the vertical portions of the head capacity curve, and submergence and capacity are self-regulating within limits. When pump is overdesigned or when actual system resistance is less than rated so that system S' obtains, then pump operation is adversely affected with increased cavitation noise and still further reduced submergence.

Method 3. (See figure 9470-36.) Considered undesirable but preferable to method 1. Purpose is to keep pump from severe cavitation at reduced loads by steepening system curve S to S', S'', etc., in proportion to load so that pump operations at points B or B'. The great objection to this

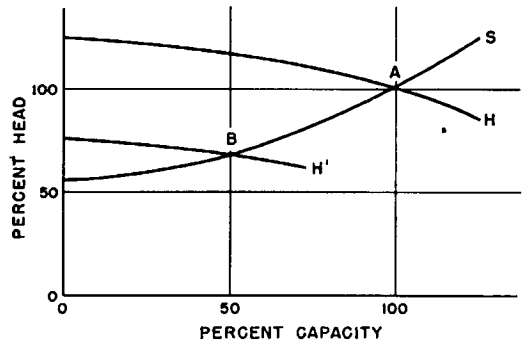


Figure 9470-34

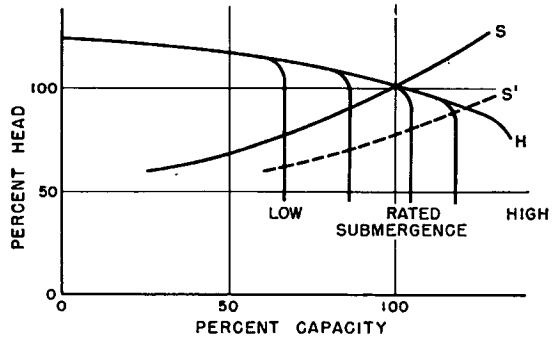


Figure 9470-35

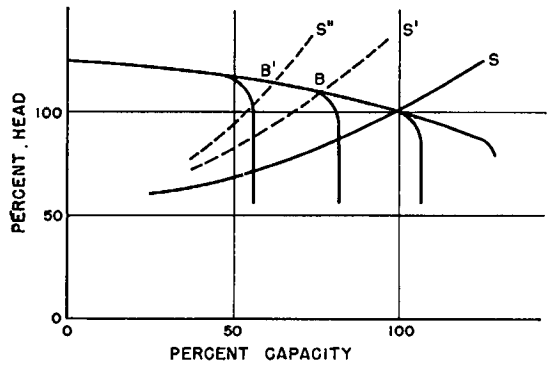


Figure 9470-36

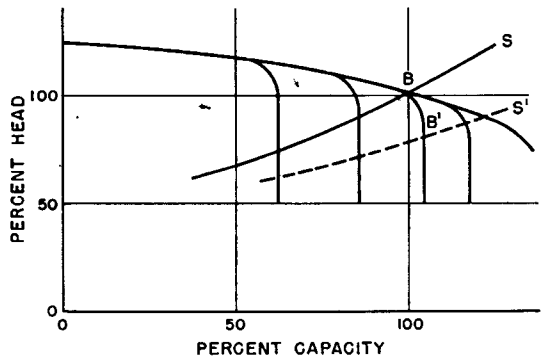


Figure 9470-37

method is the inability of the pump to handle increased capacity without manipulation of the discharge valve.

Method 4. (See figure 9470-37.) Preferred to methods 1 or 3. Increase the submergence by increasing the gross capacity of the pump. Also has the effect of cutting down on the available capacity of the pump by an amount equal to 50 to 100 percent of the quantity recirculated depending on the point in the system from which recirculation takes place. The system curve when recirculating (S') is to the right of the normal system curve S. Since the amount of recirculation is limited by the size of the recirculating line, this method is not as critical in adjustment as method 3.

Part 3. Maintenance (where 3-M system is installed, conduct preventive maintenance in accordance with 3-M cards)

9470.161 USE OF MANUFACTURERS' INSTRUCTION BOOKS

Manufacturers' instruction books are furnished for all but the simplest types of pumps. These instruction books contain detailed information concerning the operation, maintenance, and repair of the specific pumps installed. They should be studied carefully before attempting to operate or service the unit. If the instructions given in this chapter conflict with manufacturers' instruction books, the Naval Ships Systems Command should be consulted.

9470.162 OBSERVATIONS IN SERVICE

Make frequent regular inspections of pumps while operating. Investigate immediately any change in the sound of a pump. At least once an hour check:

1. Bearing temperatures.
2. Stuffing boxes.
3. Pressure gages.

9470.163 SHAFT PACKING

The packing in pump stuffing boxes should be renewed about once every two months to make sure that it does not get hard and score the shaft. When repacking a pump stuffing box, install packing of the proper length to give uniform thickness all around the shaft sleeves. Install rings with joints staggered. When installing new packing, the stuffing box should be packed loosely and the packing gland set up lightly, allowing a liberal leakage for stuffing boxes subjected to pressure above atmosphere. Then, with pump in operation, tighten the gland in steps so as to compress the packing gradually, thereby avoiding excessive heating and possible scoring of shafts or shaft sleeves. After packing has been compressed, it will usually be found necessary to add more packing. All pumps should be run with a slight drip from stuffing boxes in order to lubricate and cool the packing.

9470.164 LANTERN RINGS, SLEEVES, AND FLINGERS

1. Where a stuffing box is fitted with a lantern ring, be sure to replace the packing beyond the lantern ring at the bottom of the stuffing box and see that the sealing water connection to the lantern ring is not blanked off by the packing.

2. Usually pump shafts are fitted with sleeves in way of shaft packing. See that shaft sleeves are tight on shafts. These are usually secured by keying to shafts or by screw threads cut to the hand opposite to the shaft rotation. Care should be taken to see that water does not leak between shaft and shaft sleeves. If shafts or sleeves in way of packing are roughened or grooved, they should be turned or ground to give a smooth surface; otherwise, difficulty will be experienced in keeping tight.

3. Water flingers are always fitted on shafts outboard of stuffing box glands to prevent leakage from the stuffing box from following along the shaft and entering the bearing housings. Care should be taken to see that flingers are tight on the shafts. If flingers are fitted on the shaft sleeves instead of on the shafts, see that no water leaks under the sleeves. If leakage occurs, a fiber or asbestos washer should be fitted between the end of sleeve and the shaft shoulder or impeller nut and all clearance between shaft and sleeve filled with white or red lead to exclude water.

9470.165 IMPELLER AND CASING WEARING RINGS

The clearance between the impeller and the casing wearing rings should be maintained as shown on the manufacturers' plans. On low-pressure pumps such as circulating, flushing, condensate, and others where the drop in pressure across the wearing rings does not exceed 150 psi, the wearing ring diametral clearance may wear as much as 0.015 to 0.030 inch without appreciable effect on the capacity of the pump; hence with this class of pump wearing ring renewel should not be undertaken until wear or increased clearance has exceeded these figures. For pressures up to 75 psi, a wear of 0.030–0.050 inch is permissible; for pressure of 75 to 150 psi, a wear of 0.020 to 0.030 inch is allowable. However, the capacity of the pump should be taken into consideration since it is evident that the percentage leakage with 0.030 inch clearance is comparatively large for a small pump, but would be negligible for a large capacity pump. The engineer officer should decide, in each case, whether or not the amount of wear of clearance rings dictates renewal. For high pressure boiler feed pumps, however, the effect of increased wearing rings clearance makes itself felt by reduction in the efficiency and in the maximum capacity of the pump. For this class of service, the wearing rings should be renewed when the clearance shown on manufacturers' plans is exceeded by more than about 100 percent. However, in no case is it considered that wearing rings need to be renewed unless the wear amounts to 0.015 inch.

9470.166 LUBRICATION

1. Before refilling a forced lubrication system with clean oil, the system should be thoroughly flushed out with hot

oil, Symbol 2075 or 2110, by pumping such flushing oil through the system. Immediately after flushing, the flushing oil should be drained from all parts of the system, the sump tank thoroughly wiped with clean, dry rags, and the system filled with clean lubricating oil which should then be pumped through the system for a few minutes. Where flushing oil is not available, a light grade of lubricating oil cut with an equal quantity of kerosene may be used; however, it should be noted that when the kerosene mixture is used, there is danger of damage to gears and oil pump if the unit is run at high speed, particularly if a worm gear drive is used.

2. The oil reservoir should be checked frequently to ensure that it is free of water. If water is found in the oil, check carefully for leakage from pump and turbine glands and leaky oil cooler. Clean the oil filters or strainers frequently; with edge filtration (Cuno or similar) type filter, drain out sludge and sediment which occasionally accumulates in bottom of filter when the unit is not running.

3. Housings of grease lubricated ball bearings should be checked occasionally to ensure that the housing is free of water and foreign matter. At the same time check to ensure that bearings are properly locked on the shafts. If water is found in the bearing housing, it should be traced to its source. It is usually caused by leakage from pump packing. Effective means should be taken to prevent water from entering the housing. The frequency with which grease must be added depends on the service of the machine and the tightness of the housing seals, and should be determined for each machine by the engineer officer. Ordinarily, the addition of grease will not be necessary more often than once in six months. The use of excessive quantities of grease is to be avoided because it has been a major cause of bearing failures. When a bearing housing is too full of lubricant, the churning of the grease generates heat which in turn causes deterioration of the grease. Under these conditions the grease separates into oil and minute abrasive particles, becomes increasingly sticky, and tends to seal the bearing against fresh lubricant until the resulting friction, heat, and wear cause failure of the bearing. To avoid the difficulties caused by an excessive amount of grease, grease should be added only when necessary, and, when grease is added, it should be done as follows:

a. Remove bearing drain plug, and make sure the passage is open by probing with a clean screw driver or similar implement.

b. Clean the grease cup thoroughly. Install bottom portion of grease cup on the grease pipe. Put in the top portion of the grease cup no more grease than will half fill it.

c. Empty and clean out the receptacle of the grease fitting down to the neck, then fill with clean grease.

d. Replace the grease cup and screw it down as far as it will go. The purpose of screwing the grease cups down is to protect the machine from being over-greased due to accidental or unauthorized turning of the grease cup.

e. Run the machine and let grease run out of drain hole until drainage stops (normally about 30 minutes). Replace the drain plug.

4. Oil in the lubricating-oil system should be renewed whenever it becomes emulsified or foams excessively, or when it contains sludge, dirt, or foreign matter.

9470.167 BEARINGS, OIL-LUBRICATED

Check bearing clearance with leads, and take bridge gage readings (where bridge gages are furnished) of all sleeve- or shell-type bearings at least every six months. Clearance should be maintained as shown on manufacturers' plans. In the absence of information on plans, the instructions in chapter 9005, table B-4, should be followed.

9470.168 THRUST BEARINGS

Thrust bearings should be examined quarterly and the position of rotors checked. When checking rotor position due allowance should be made for expansion of shaft from cold condition to hot running condition.

9470.169 INTERNAL WATER-LUBRICATED BEARINGS

1. Most designs of vertical condensate main circulating pumps are fitted with an internal water-lubricated bearing inside the pump casing. Where this type of bearing is fitted in a pump, it is of particular importance that an adequate supply of clean water for lubricating and cooling be supplied to the bearing. A number of different materials have been used for internal water-lubricated bearings including:

a. Laminated phenolic material grade FBM (fabric base bakelite or micarta).

b. High lead content bronze.

c. Graphited bronze.

d. Lignum vitae bearings are also satisfactory; however, due to greater difficulty of installation, its use except in an emergency, is not recommended.

2. The condition of all types of internal water-lubricated bearings should be checked frequently to guard against excessive wear which would cause misalinement and possibly shaft failure.

9470.170 COOLING WATER

A liberal supply of clean cooling water, usually sea water should be supplied to the oil coolers, or to the oil cooling coils, of all units fitted with oil cooling arrangements. A frequent check of oil temperatures should be made to ensure that the cooling system is functioning satisfactorily. This is particularly important on high speed boiler feed pumps and on turbine-driven units fitted with worm gear type of reduction gear.

9470.171 CARE OF DRIVING UNITS

Instructions for the maintenance of driving engines, turbines, motors, and controllers will be found in other chapters of this manual and should be followed.

9470.172 COUPLINGS

1. When the driving unit is connected to the pump by means of a flexible coupling, it should be borne in mind that flexible couplings are intended to take care of but slight misalinement. Where misalinement is small, the coupling should operate satisfactorily without requiring frequent renewal of coupling parts; but if misalinement is excessive, the coupling parts are subjected to severe punishment, necessitating frequent renewal of pins, bushings and bearings.

2. Couplings with self-contained oil have been in use over a period of years and if kept lubricated have proved satisfactory. However, some couplings fail due to loss of

oil or because no oil has been added, resulting in wear and breakage of teeth. It is therefore recommended that the following precautions be observed:

a. Inspect the flexible coupling monthly by removing filler plug to ascertain that there is an adequate supply of lubricant.

b. Whenever coupling is dismantled, inspect teeth to see that they are in good condition. Care should be taken to assure that coupling is reassembled to suit match marks. When reassembling coupling, check alinement of turbine and pump to prevent excessive coupling wear.

9470.173 ALINEMENT

Shaft alinement should be checked frequently and if it is found that shafts are out of line or inclined at an angle to each other, realinement of the unit should be undertaken in order to avoid shaft breakages and renewal of bearings, pump casing wearing rings, and throat bushings. Whenever practical, the alinement should be checked with all piping in place, all tanks and piping filled. Due allowance should be made for change in position of parts from the cold-check condition to the hot-operating condition. Such allowance for change in alinement caused by changes in temperature is assuming an ever-increasing importance owing to the higher steam, exhaust, and water temperatures now used. The necessity for allowing for expansion of parts cannot be overstressed.

9470.174 SPEED-REGULATING, SPEED-LIMITING GOVERNORS

All turbine-driven units are fitted with a speed-regulating or speed-limiting governor. The governor should be set to give rated speed at rated load conditions. With the governor set as above, the turbine speed should not exceed the rated speed by more than five percent for any condition of load, including shutoff of the pump. If the governor will not function within the above limit, it should be overhauled and the cause of faulty operation located and remedied. For satisfactory parallel and series operation of feed booster pumps and boiler feed pumps, it is essential that the governors of all identical pumps be set for the same speed.

9470.175 PUMP PRESSURE GOVERNORS

In addition to the governors previously mentioned, turbine-driven boiler feed pumps and fire pumps are fitted with pump pressure regulating governors. (See section IV.)

9470.176 TESTS AND INSPECTIONS (CENTRIFUGAL PUMPS)

1. **Weekly.**

a. Operate all pumps by steam or power. If power is not available, move by hand.

b. Lubricate the pressure regulating governor, if of a type requiring lubrication.

c. Check lubricating oil for water and for condition of oil by sampling.

2. **Quarterly.**

a. Check thrust position of pump rotor.

b. Sound and set up as necessary all foundation bolts and secure all foundation dowel pins.

3. **Annually.**

Check bearing clearances by leads or crown thickness measurements.

4. **Overhaul cycle.**

a. Check internal water lubricated bearings and shafts in way of same for wear and scoring.

b. Open pump and reduction gear casings; inspect and clean.

c. Check clearances of all diaphragms and casing throat bushings, impeller and casing wearing rings, pressure break-down drums and bushings, and renew as necessary.

NOTE: These tests and inspections are the minimum necessary to give adequate assurance of safe reliable operation of equipment. Indications of low discharge pressure or other manifestations of improper operation should indicate more frequent or extensive tests and inspections. Salt water and brine pumps should be opened and inspected more often as performance indicates. In general, pumps should be turned over and lube oil checked each time a unit is placed in service. Cleaning of lube oil sumps should be dictated by condition of oil. The degree of oil contamination will dictate the necessity for further inspection of bearings and journals. For tests and inspections of turbines, see chapter 9500, article 9500.47.

Part 4. Repairs

9470.181 ASSEMBLY DRAWINGS

When repairing or making an interior examination of a pump, it is essential that all drawings and available dimensional data relative thereto be at hand. Frequently, important dimensions become altered such as bridge gage readings of bearings, clearance between impeller and casing wearing rings, water seal or gland adjustment. This results in poor operation, which will naturally continue in spite of other major repairs unless the real cause of the trouble is rectified.

9470.182 RENEWAL OF WEARING PARTS

The parts of a centrifugal pump most frequently requiring repair or replacement are:

1. **Casing rings and impeller rings.** As the purpose of these is to keep the internal bypassing of the liquid to a minimum, the clearances should be checked periodically and whenever the pump casing is opened up.

2. **Shaft sleeves.** Operating personnel frequently take up too hard on the packing in at attempt to stop stuffing box leakage. This causes scoring of shaft sleeves. Sleeves should be examined whenever pump is opened up. If not badly scored, they should be smoothed up. When worn, they should be replaced.

3. **Bearings.** Worn sleeve bearings cause the rotor to drop, which causes wear of casing and impeller rings. Bearings of centrifugal pumps should be rebabbitted in accordance with the table in chapter 9400, when bridge gage readings or leads show that maximum allowable wear has occured. The oil clearances for bearings will be found on the manufacturers' plans or instruction book. However, in the absence of such data, the table of tolerances and clearances, chapter 9400, should be followed.

4. **Bushings.** Bushing clearances should be measured whenever the pump is opened up. Bearing wear will probably cause bushing wear, and bushing should be renewed provided bearings are restored to their original readings.

9470.183 EMERGENCY REBABBITTING OF SMALL BEARINGS

To rebabbitt a small bearing in an emergency, coat the journal heavily with banana oil and, with the shaft properly centered in the bearing housing, pour the babbitt around the journal. The bearing will break clean from the journal and can then be used with little or no scraping in.

9470.184 CENTERING OF IMPELLERS

It is essential to the best operation of a centrifugal pump that the center of the impeller exit passages line up accurately with the center of casing or diffuser waterways. Otherwise excessive turbulence and friction losses may obtain and thrust load increased.

9470.185 ALINING PUMP

Pumps should be periodically checked for alinement since piping strains may develop distorting the unit. Bearing wear and distortion of supporting structure may also contribute to misalinement. In general flexible and rigid coupled units may be lined up by disconnecting couplings and checking rims by straight edge and faces for parallelism by feeler gage. Other types of pumps should be alined in accordance with manufacturers' instructions.

9470.186 BALANCING

All pump and driving unit rotating parts are balanced dynamically for all speeds from 0 to 125 percent of rated speed. The parts are usually balanced on a balancing machine to ensure accurate balance. Balancing machines are usually available only at naval shipyards. A portable balancing outfit (Davey vibrometer) is available on tenders for use of individual ships. With a portable unit, the pump and turbine rotors may be balanced in place. Whenever possible the balancing machine or portable balancing outfit should be used in preference to any other method of balancing.

Part 5. Safety Precautions

9470.191 PRECAUTIONARY INSTRUCTIONS

1. Test relief valves, where fitted, to function at the designated pressure.
2. Never attempt to jack a pump by hand while the steam valve to the pump is open.
3. Boiler feed pumps and other pumps in the feed system shall not be used for purposes other than those connected with boiler of feed water service, except in an emergency.
4. Do not tie down or otherwise render inoperative the overspeed trip, speed-limiting or speed-regulating governor.
5. See that overspeed trips, where fitted, are set to shut off steam to the unit when rated speed is exceeded by 10 percent.
6. See that speed-limiting and speed-regulating governors are set to limit the speed of the unit to rated speed under rated conditions, and that rated speed is not exceeded by more than five percent for any condition of loading.
7. Check the setting of overspeed trip and speed-limiting governors at least quarterly.
8. When inspecting or overhauling vertical rotary pumps never rely on a flexible coupling to support the total weight of the pump rotors. Always support the pump rotors by means of wire rope slings, block and tackle, or chocks before working on the pump.

SECTION III. ROTARY PUMPS

Part 1. General

9470.201 INTRODUCTION

Positive displacement rotary pumps have largely supplanted reciprocating pumps for pumping all viscous liquids on naval ships. Rotary pumps are also used for gasoline or other low viscosity volative liquids where a high suction lift is required and the service is intermittent. Rotary pumps are not suitable for handling corrosive liquids nor liquids containing abrasive solids. In general, rotary pumps are self-priming.

9470.202 DESCRIPTION

Rotary pumps consist of a casing containing a rotating assembly running in close clearances and in such manner that liquid is trapped in a moving space formed by the rotor and casing and carried to the discharge side where the liquid is pushed or squeezed from the pumping space to the discharge. The rotor assembly may consist of pairs of intermeshing gears, screws, vanes, lobes, and cams; or sliding pistons, vanes, blocks, or other forms. Those more common to the naval service are described below.

9470.203 CLASSIFICATION AND TYPES

Rotary pumps are classified as to:

1. Service, application.
2. Manufacturers' name or trade name.

This method of identification is indeed more in common use for rotary pumps than for other types of pumps or even other equipment. Most manufacturers specialize in only one or two types and these have significant design characteristics.

3. Type rotor.
   a. Cam and plunger (figure 9470-38).
   b. Screws with timing gears (figures 9470-39 and 9470-40).
   c. Screws without timing gears (figure 9470-41).
   d. Impellers (cams, lobes, or special tooth forms) with timing gears (figures 9470-42 and 9470-43).
   e. Gears (spur helical, herringbone) (figure 9470-44).
   f. Gears with separate timing gears (figure 9470-45).
   g. Axial or radial piston variable displacement (figure 9470-46).
   h. Internal gears (figure 9470-47).
   i. Vanes (figures 9470-48 and 9470-49).
4. Shaft position.
   a. Horizontal.
   b. Vertical.
   c. Inclined.
5. Type of drive and power.

9470.204 SERVICE CLASSIFICATION

Rotary pumps are used in the naval service for the following applications:

1. Boiler fuel oil service.
   a. Main.
   b. Port and cruising.
2. Fuel oil booster and transfer service.
3. Diesel oil service and transfer.
4. Fuel oil tank drain (stripping).
5. Lubricating oil service and transfer.
6. Fuel oil, gasoline, or water cargo.
7. Cargo stripping.

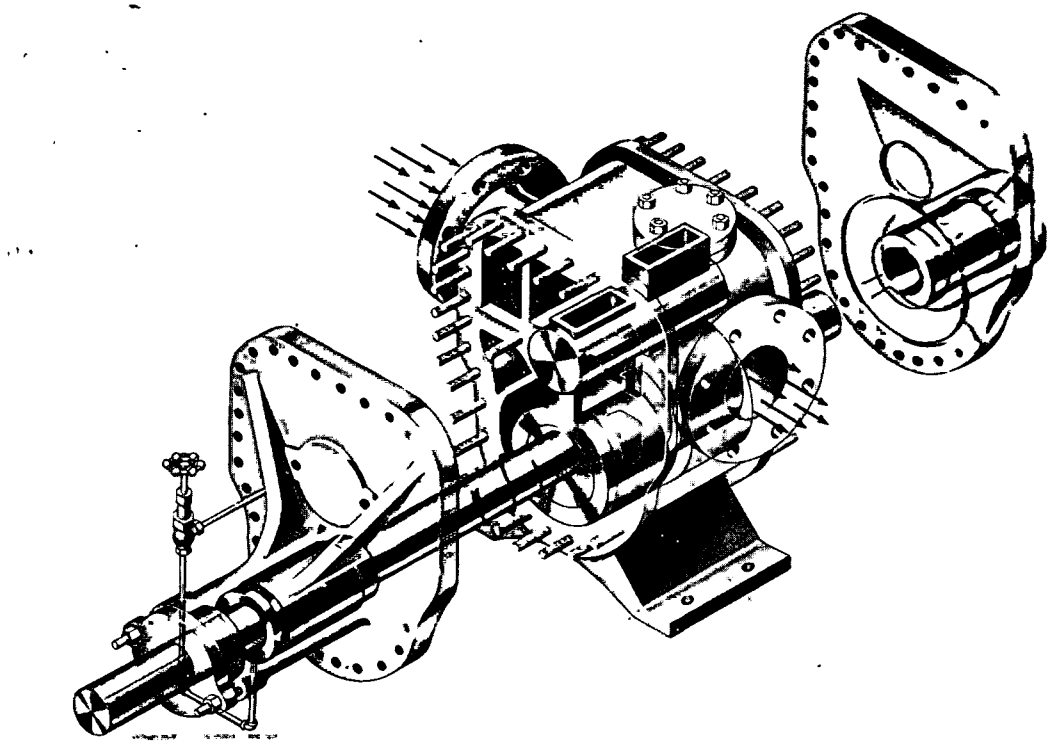


Figure 9470-38

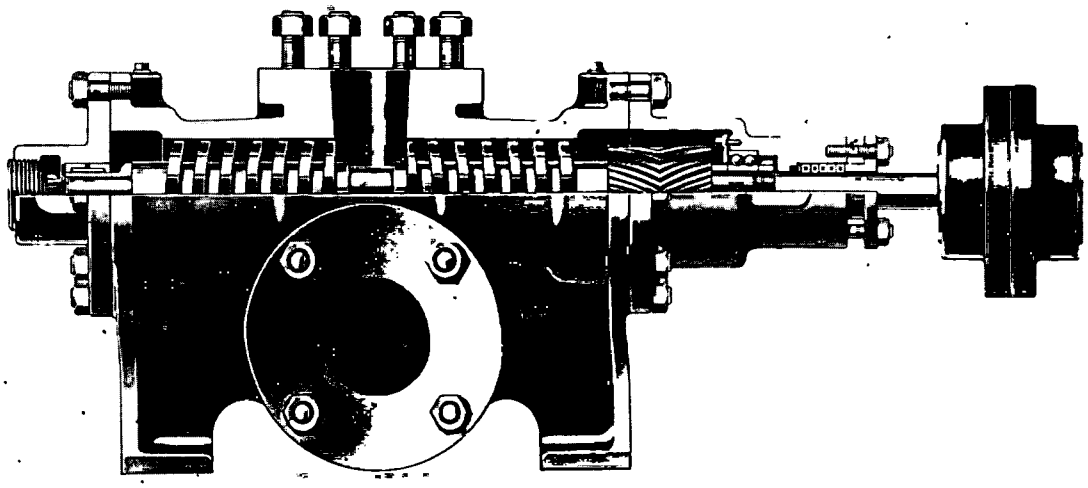
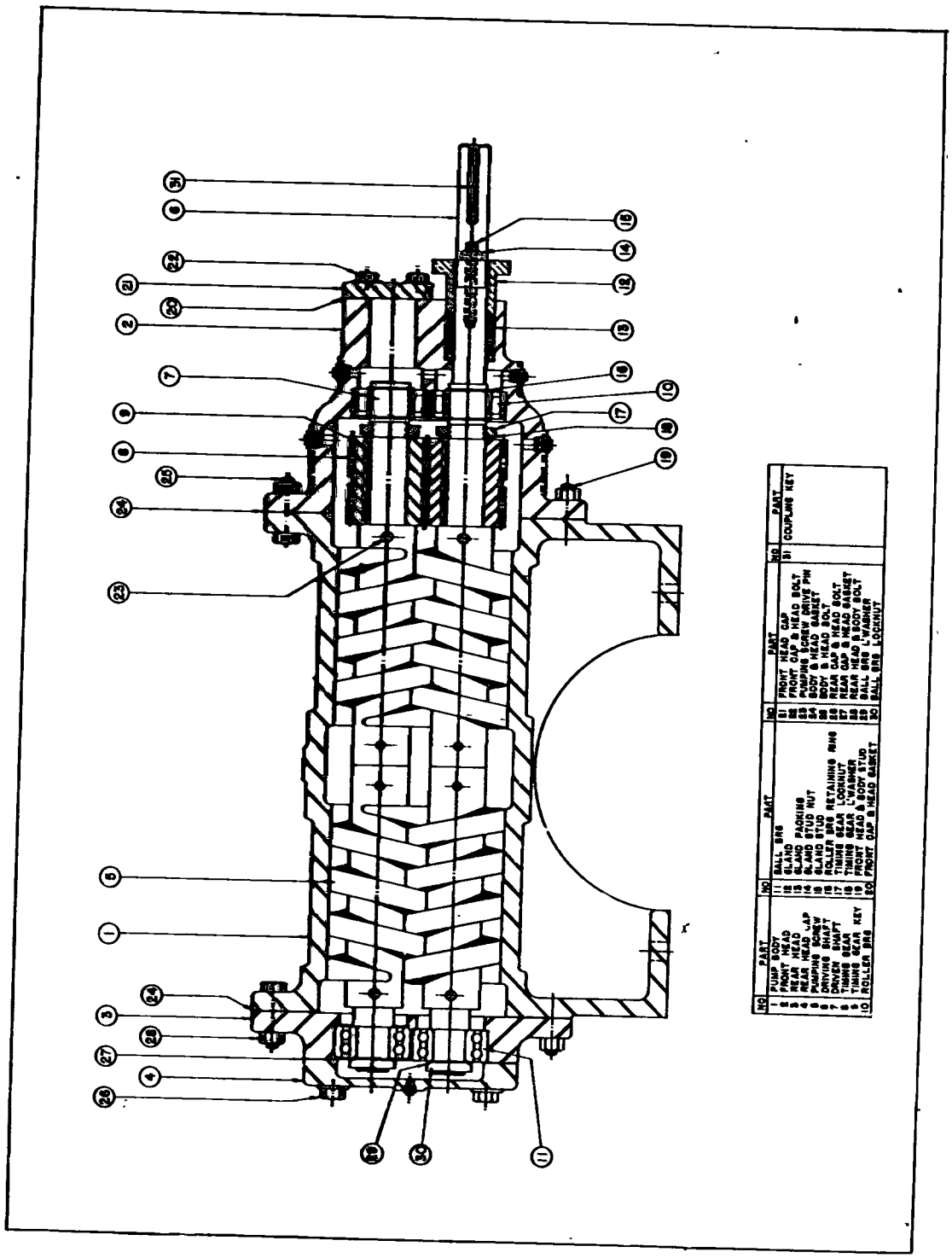


Figure 9470-39



NO	PART	NO	PART	NO	PART
1	PUMP BODY	11	BALL BRG	21	FRONT HEAD CAP
2	FRONT HEAD	12	GLAND	22	FRONT CAP & HEAD BOLT
3	REAR HEAD	13	GLAND PACKING	23	PUMPING SCREW & DRIVE PIN
4	REAR HEAD CAP	14	GLAND STUD NUT	24	BODY & HEAD GASKET
5	PUMPING SCREW	15	GLAND STUD	25	BODY & HEAD BOLT
6	DRIVING SHAFT	16	ROLLER BRG RETAINING RING	26	REAR CAP & HEAD BOLT
7	DRIVEN SHAFT	17	TIMING GEAR LOCKNUT	27	REAR CAP & HEAD GASKET
8	TIMING GEAR	18	TIMING GEAR LWASHER	28	REAR HEAD & BODY BOLT
9	TIMING GEAR KEY	19	FRONT HEAD & BODY STUD	29	BALL BRG HOUSING
10	ROLLER BRG	20	FRONT CAP & HEAD GASKET	30	BALL BRG LOCKNUT
				31	COUPLING KEY

Figure 9470-40

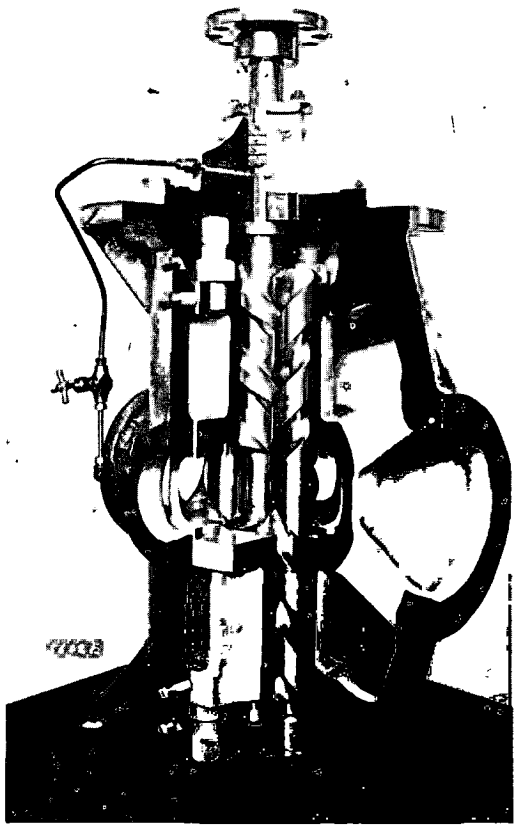


Figure 9470-41

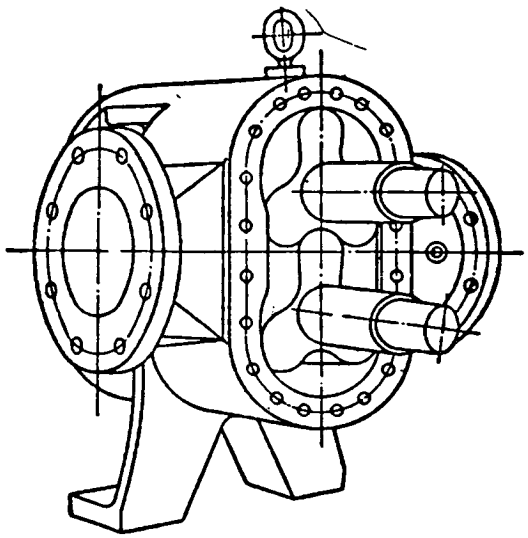


Figure 9470-42

9470.205 GENERAL CHARACTERISTICS

Rotary pumps are of the positive displacement type and their capacity is directly proportional to the pump speed subject to losses described below. The pressure developed by rotary pumps is that imposed by the resistance of the system to which it discharges and is limited only by the bursting strength of the pump or power available. For this reason, relief valves are generally fitted on the pump discharge.

9470.206 PUMP DISPLACEMENT

This displacement of a rotary pump is the volume that the rotating elements displace per revolution of the rotor shaft. Displacement is independent of all operating conditions and is a theoretical capacity assuming complete filling of the pumping spaces and no losses.

9470.207 CAPACITY

The capacity of a rotary pump is the quantity of liquid actually delivered under specified conditions. It is equal to the displacement times the revolutions per minute minus losses due to slip or inlet conditions. The ratio of the actual capacity to the displacement expressed in percent is known as the volumetric efficiency.

9470.208 SLIP

Slip is the quantity of liquid that bypasses from discharge to suction through the internal working clearances of the pump. Slip increases with pressure and inversely with viscosity.

9470.209 INLET CONDITIONS

The conditions of the liquid at the suction of the pump are most important to its performance. These factors include suction pressure or lift, viscosity, vapor pressure at operating temperature, amount of entrained of dissolved air or gas. Capacity losses due to inlet conditions are caused by incomplete filling of the pumping element and are not considered as slip.

9470.210 EFFECT OF HIGH SUCTION LIFT

Investigations conducted by the Naval Ship Systems Command have revealed that rotary pumps will lift oil equally as well if not better than a reciprocating pump in average condition or repair. However, the volumetric efficiency will drop off rapidly if the pump is required to operate on a vacuum or suction lift exceeding that for which it is designed. This is clearly indicated by figure 9470-50. The upper curve A on this figure shows that with the pump handling viscous oil (700 SSF) the capacity is maintained well above rated capacity of 250 gallons per minute at 25 inches of mercury. However, when the pump is handling the same oil at a lower viscosity (higher temperature) (70 SSF), the capacity drops off abruptly over 19 to 20 inches of mercury, suction lift. This reduction in capacity is only partly due to increased slip and to a great extent to the effect of deaeration and vaporization of the oil.

9470.211 EFFECT OF ENTRAINED OR DISSOLVED GASES

1. Most liquids will contain a varying amount of air or gas by entrainment or in solution. If the suction lift is high, entrained gas will expand and dissolved gas will come

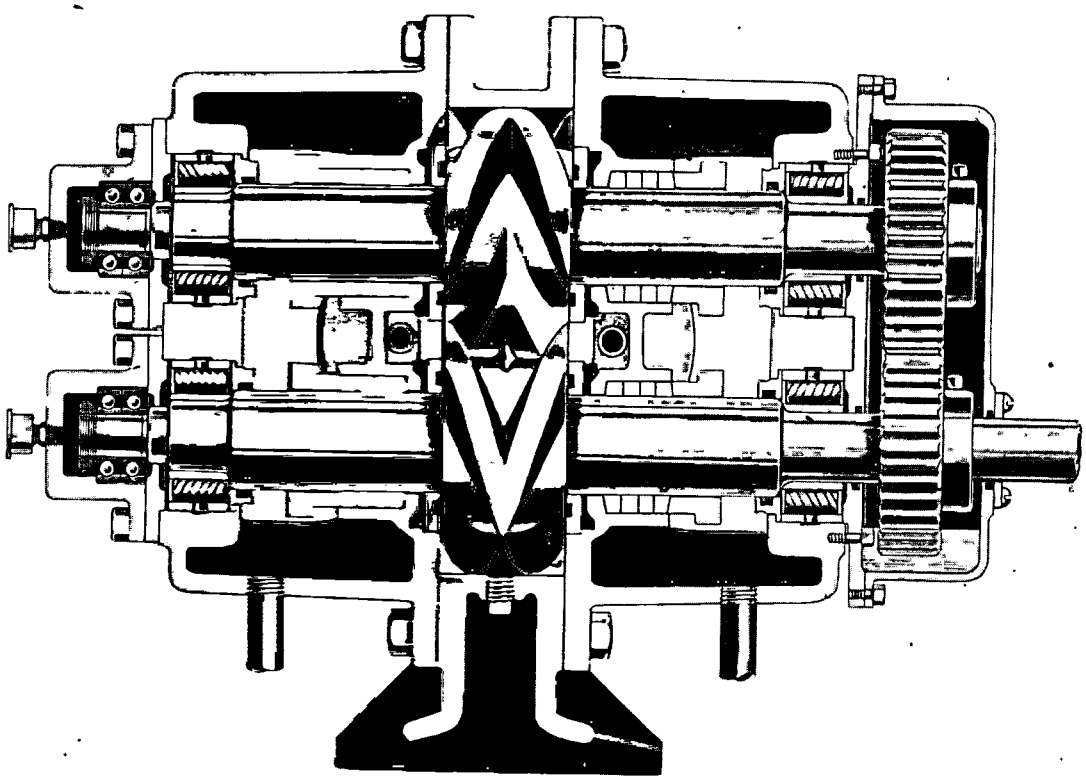


Figure 9470-43

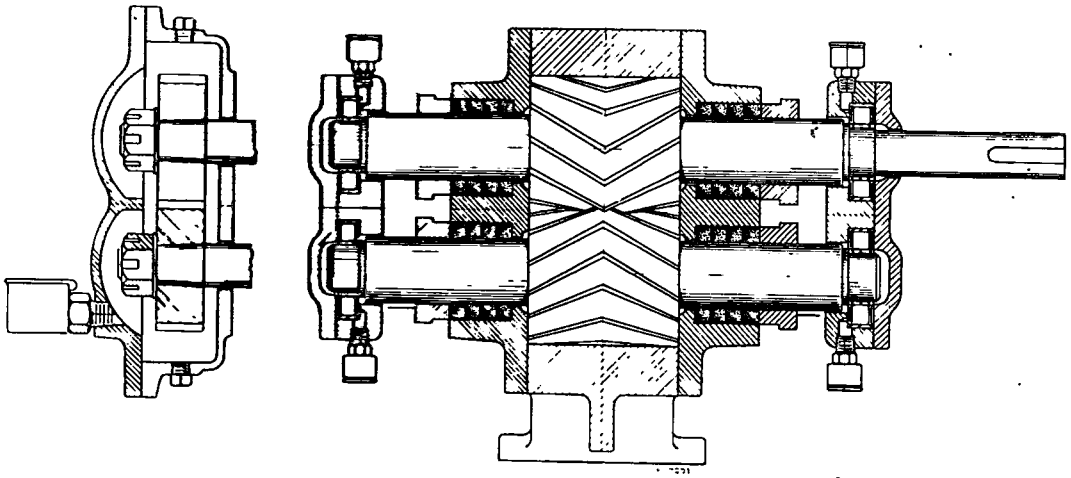


Figure 9470-44

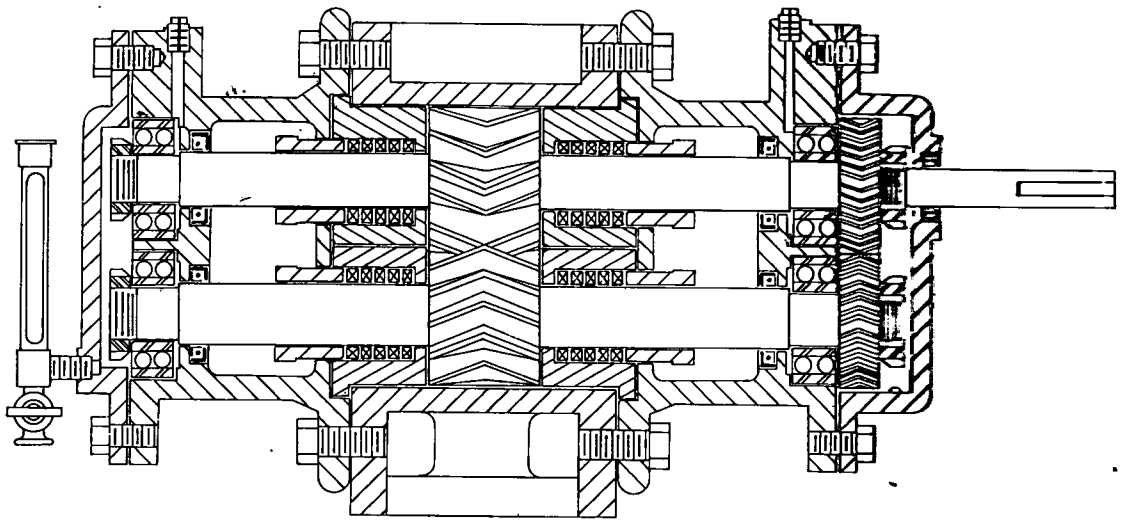


Figure 9470-45

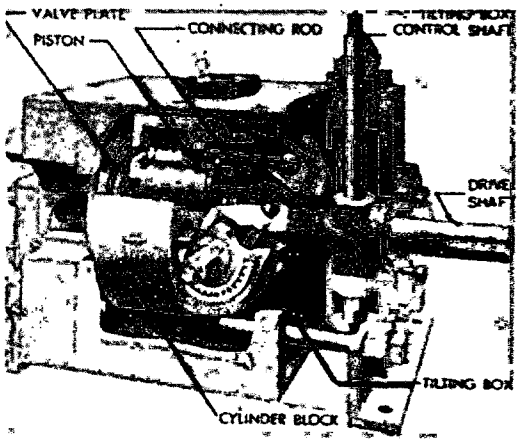


Figure 9470-46

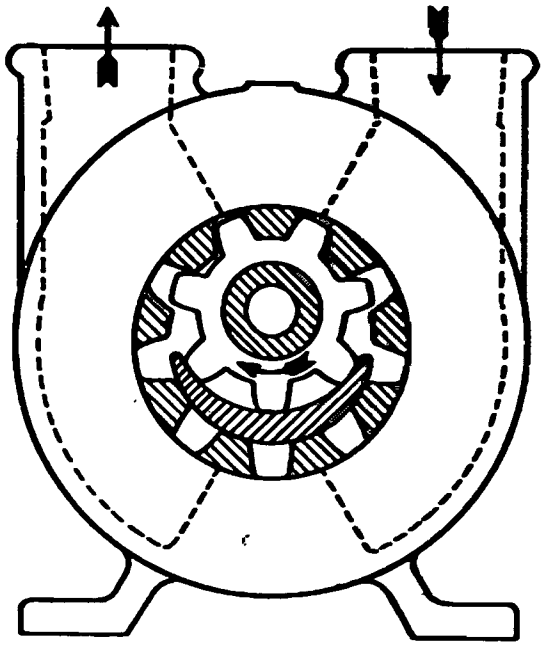


Figure 9470-47

out of solution. This reduces the pump's liquid capacity and will result in noise, vibration, and pressure pulsations. Under some conditions these pressure pulsations can attain a high magnitude which will cause damage to piping or to the pump itself. The magnitude of these pulsations is dependent on pressure pump speed and number of closures of the rotor element per revolution.

2. **Entrained gases.** The amount of entrained air or gas contained in the liquid handled varies greatly and depends on the viscosity, type of liquid, the extent of agitation and exposure, and the type of system. Main lubricating oil systems are highly susceptible to air entrainment and foaming. Figure 9470-51 shows the reduction in liquid capacity plotted against suction lift for different amounts of air or gas entrained.

3. **Dissolved gases.** The amount of air or gas contained in a liquid by solution varies with the type of liquid, and the time and conditions of exposure. Gasoline may dissolve up to 20 percent of air by volume; lubricating oil may contain 10 percent by volume. These figures are approximate. Figure 9470-52 shows the reduction in liquid capacity plotted against suction lift for different amounts of air or gas in solution. Note that whereas the amount of entrained air affects the capacity even at 0 suction lift, the dissolved air remains in solution until the pressure is lowered.

9470.212 EFFECT OF VISCOSITY

Rotary pumps are suitable for handling liquids over a wide range of viscosities. The power required increases

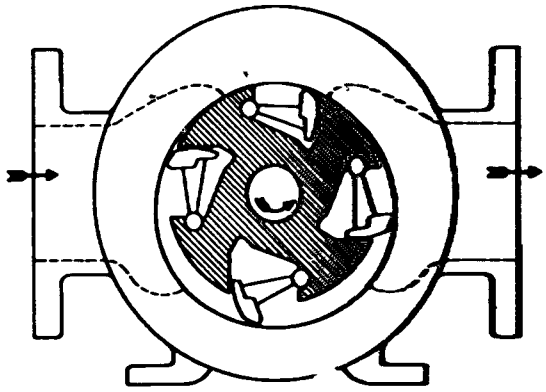


Figure 9470-48

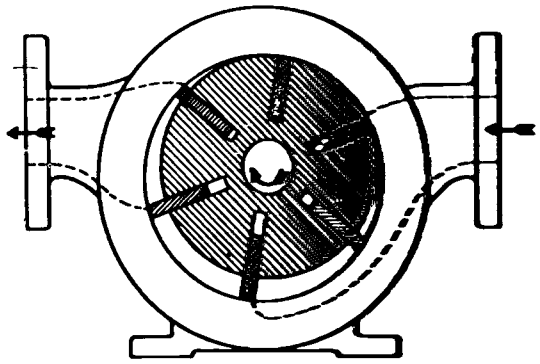


Figure 9470-49

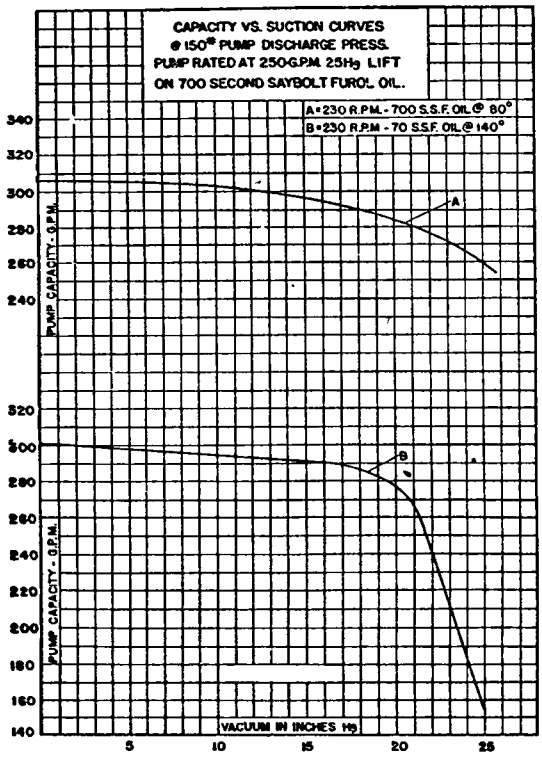

CAPACITY VS. SUCTION CURVES
@ 150# PUMP DISCHARGE PRESS.
PUMP RATED AT 250 G.P.M. 25 Hg LIFT
ON 700 SECOND SAYBOLT FUROL OIL.
A = 230 R.P.M. - 700 S.S.F. OIL @ 80°
B = 230 R.P.M. - 70 S.S.F. OIL @ 140°
PUMP CAPACITY - G.P.M.
VACUUM IN INCHES Hg
A
B

Figure 9470-50

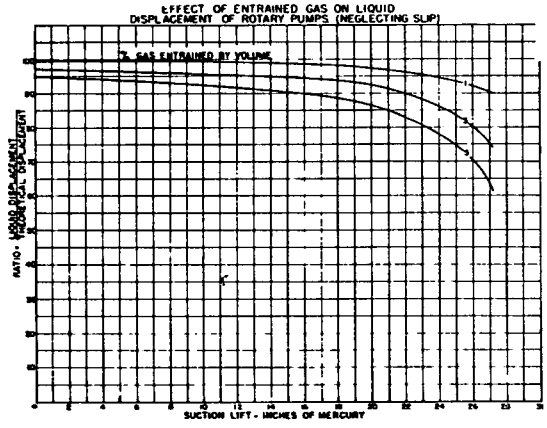

EFFECT OF ENTRAINED GAS ON LIQUID
DISPLACEMENT OF ROTARY PUMPS (NEGLECTING SLIP)
% GAS ENTRAINED BY VOLUME
SUCTION LIFT - INCHES OF MERCURY

Figure 9470-51

with viscosity due to increased friction within the pump. The speed of the pump should be reduced when handling increased viscosity in order to allow the rotors time to fill. The Naval Ships Systems Command has made extensive tests on the maximum rubbing or pitch line speed for different oils and these limitations are incorporated into specifications for rotary pumps for naval service.

9470.213 MAIN FUEL OIL SERVICE

For boiler fuel oil service, screw type pumps, figures 9470-39, 9470-40, and 9470-41, are extensively used in naval service.

To a lesser extent, the Kinney rotary pump, figure 9470-38, and gear pumps, figure 9470-44, are used. Other types, such as the three-lobe, figure 9470-42, and the heliquad, figure 9470-43, are not inherently suited for high pressure fuel oil service due to their larger diameter rotors, which cause excessive shaft deflection and heavy bearing loads. Pumps for this service are designed for suction conditions up to 20 inches suction lift and for discharge pressures of 350 or 1050 pounds per square inch gage depending on the burner system for the particular ship on which they are installed.

9470.214 PORT AND CRUISING FUEL OIL SERVICE

Two types of port and cruising fuel oil service pumps are in current use. Some ships have installed motor-driven, variable delivery, axial piston type pumps similar to figure 9470-46. This type is intended to operate with varying capacity at constant motor and pump speed. This is accomplished by means of a constant pressure governor which varies the capacity of the pump through a stroke adjusting mechanism in response to burner demands. This control

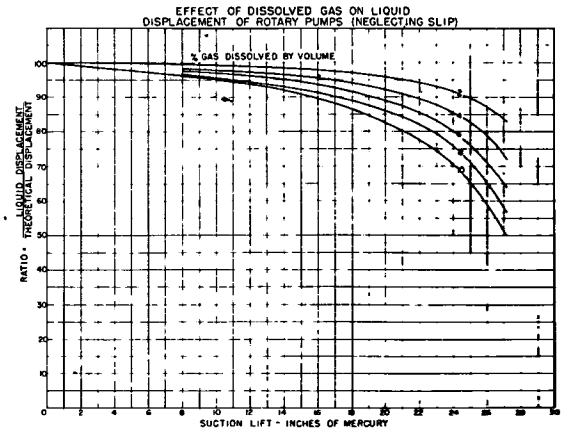

EFFECT OF DISSOLVED GAS ON LIQUID
DISPLACEMENT OF ROTARY PUMPS (NEGLECTING SLIP)
% GAS DISSOLVED BY VOLUME
RATIO - LIQUID DISPLACEMENT / THEORETICAL DISPLACEMENT
SUCTION LIFT - INCHES OF MERCURY

Figure 9470-52

is usually augmented by a manually-operated capacity control by means of which the pressure control can be made inoperative and the capacity adjusted by hand to meet the demand. The majority of ships, including those of latest design, are equipped with pumps of a constant delivery type, similar to those for main fuel oil service, driven by two-speed motor and capacity further adjusted by means of a bypass valve, which returns the unused portion of the oil from the burner line to the pump suction.

9470.215 FUEL OIL BOOSTER AND TRANSFER PUMPS

1. Fuel oil booster and transfer pumps most extensively used in the service are:
   a. The Kinney cam and plunger (figure 9470-38).
   b. The DeLaval IMO pump (figure 9470-41).
   c. The Quimby or Sier-Bath screw pump (figures 9470-39 and 9470-40).
   d. The Northern cycloidal (figure 9470-42).
   e. The Kinney heliquad (figure 9470-43).
2. Fuel oil booster and transfer pumps encounter severe operating conditions. They are required to pump oil through long lengths of suction lines which may have a number of pockets or traps. The oil may have a high viscosity which results in high friction loss with high net suction lift at the pump inlet. For this reason, such pumps are usually driven by multispeed motors or steam turbines to ensure a wide range of capacities to avoid operating at too high a suction lift with cold oil.

9470.216 DIESEL OIL SERVICE

This is not difficult service, and most pumps described in preceding paragraphs are suitable. For economy of space and weight, those shown in figures 9470-44, 9470-45, 9470-47, 9470-48, and 9470-49 are usually applied to these services.

9470.217 FUEL OIL TANK DRAIN PUMPS

Fuel oil tank drain pumps are used for stripping fuel oil tanks and must handle mixtures of oil, sludge, salt water, and sediment. These pumps are usually Northern "Cycloidal," figure 9470-42, or Kinney "Heliquad," figure 9470-43. These pumps have timed rotors which prevent metal-to-metal contact of the rotors and is a necessary feature for pumps handling abrasive fluids such as encountered in this application.

9470.218 LUBRICATING OIL

Main lubricating oil pumps for surface ships, particularly those of recent construction, have been subjected to severe operating conditions due to entrained air of the lubricating oil. This has been particularly noticeable in the case of constant speed, motor-driven, lubricating oil pumps which must run at maximum capacity regardless of the operating conditions or demands of the system. The principal effect of pumping these air-oil mixtures has been pulsating discharge pressures causing vibration of the pump and pipe, and in extreme cases breakage of lines. Action which has been taken to reduce the effect of these pulsations has included reduction of suction lift by increasing size of suction lines, and improvement of entrance conditions in the pump suction, elimination of dead end relief valve lines, and greater use of turbine-driven pumps in new construction. Pumps shown in figures 9470-39, 9470-40, and 9470-41 are used for this service.

9470.219 CARGO

Main cargo pumps are frequently of centrifugal type in view of the weight and space saving possible in the large capacity pumps now required in tanker service. For moderate and small ships, pumps shown in figures 9470-42 and 9470-43 are used.

9470.220 CARGO STRIPPING

For cargo stripping, service pumps must maintain a suction lift, or act as continuous priming pumps for centrifugal main cargo pumps. As the liquid level in the tank begins to get low, pumps, figures 9470-42, 9470-43, 9470-45, and 9470-48 are used for this service.

9470.221 BEARINGS

Rotary pump bearings are classified as either "external" or "internal" type. External bearings are separated from the liquid pumped by shaft packing or mechanical seals. Internal bearings are located between the rotor elements and the packing and depend on the liquid pumped for lubrication. Internal bearings are not suitable for pumps handling liquids containing abrasives or having low viscosity lacking in lubricating qualities.

9470.222 GLAND SEALING

Where pumps are required to take suction at lower than atmospheric pressure, it is important that the shaft stuffing boxes be fitted with a lantern ring and sealed by the liquid being pumped to prevent the entrance of air into the pump. Stuffing box gland sealing liquid is taken from the discharge of the pump.

Part 2. Operation

9470.251 USE OF MANUFACTURERS' INSTRUCTION BOOKS

The instructions contained in this chapter for operation, maintenance, and repair of pumps are general for all makes and types. For all pump applications except some small miscellaneous service motor-driven pumps, manufacturers'

instruction books are furnished with the plans. These instruction books contain detailed information concerning the specific pump installed. If any conflict exists between the instructions given in this chapter and the manufacturers' instruction book, NAVSHIPS should be consulted.

9470.252 PREPARING PUMP FOR OPERATION

Before starting a rotary pump for the first time, proceed as follows:

1. Clean all external surfaces of the pump and driver.
2. Check the auxiliary piping such as lubrication lines, gland sealing connections, water supply to oil coolers, etc.
3. Examine bearings to ensure they are properly lubricated.
4. Make sure stuffing boxes are properly packed.
5. Stuffing box glands should not be drawn too tight.
6. Pump should be primed from the discharge side using the same liquid as that to be pumped. This is necessary to lubricate the rotating elements and to seal the clearances.
7. Discharge side of the pump should be vented during the initial starting so as to be free to discharge the air from the pipe system.

9470.253 STARTING

To start a rotary pump proceed as follows:

1. Check lubricating oil level in sump or bearing housing. Fill oil cups or reservoir if fitted. If external supply of lubrication is provided, open and adjust all delivery and return valves.
2. Open steam and exhaust casing drains on turbine.
3. Open valves to pump packing gland seals, where fitted.
4. Open steam and exhaust root valves.
5. Open turbine exhaust valve.
6. Lift all relief and sentinel valves by hand.
7. Open pump discharge valve.
8. Open pump suction valve.
9. a. For turbine-driven pumps, crack turbine throttle valve sufficiently to free lines, steam chest, and exhaust casing of condensate.

   b. As soon as turbine is free of condensate, close all drains, then open throttle slowly and bring unit up to speed.

   c. For motor-driven pumps start motor. (See instructions for operation of motors and controllers, chapter 9630.)
10. Check lubricating system to see that all bearings are supplied with oil and that oil pressure is correct.
11. Check all gages to see that proper pressures are being developed.
12. For turbine-driven pumps, check revolutions per minute to see that rated speed is not exceeded by more then five percent.
13. Adjust pump shaft packing glands and gland sealing needle valve when fitted.
14. Where unit is fitted with a constant pressure governor, all the preceding operations shall be performed with the governor bypassed or the governor valve pull-open device wide open.
15. The pressure governor, if it is to be used, should now be put in operation by opening the valves in the governor actuating lines between the discharge of the pump and the governor and closing the governor bypass or pull-open device. The governor should then be adjusted to give the required pump discharge pressure.
16. After pressure governor is in operation and is functioning properly, the throttle valve should gradually be opened wide to ensure that pressure governor takes control.

Then check all pressures, temperatures, speed, and lubrication to see that normal conditions obtain through-out. With pressure governor in control, the unit should require no further attention insofar as discharge pressure is concerned. However, the operation of the unit should be checked frequently during each watch to see that all is normal.

17. If no pressure governor is fitted, the unit will require close attention to throttling to maintain the required discharge pressure, especially if the ship is maneuvering, transferring fuel, or pumping tanks.

9470.254 STOPPING AND SECURING

To stop and secure, proceed as follows:

1. Close throttle gradually, or stop motor, being sure that the pump discharge check valve closes, to prevent a backflow through the pump being secured from another pump which may be discharging into same line.
2. Close pump discharge stop valve.
3. Close exhaust valve.
4. Close pump suction valve.
5. Open turbine drains.
6. Close all supply and return valves when unit is lubricated by a detached pump or by a main lubricating oil system.
7. Close steam and exhaust root valves.
8. Close steam drains after turbine is completely drained.
9. Close pump governor actuating line valve and open governor by-pass valve or pull-open device.
10. After the unit is secured as above, read the turbine exhaust casing pressure gage and pump suction combined pressure and vacuum gage, to ensure that steam does not leak through exhaust valves and build up pressure in the exhaust casing or turbine, or that pump casing is subjected to pressure due to leakage through pump discharge stop and check valves.

9470.255 PUMP FAILS TO DELIVER LIQUID

If pump does not discharge liquid when first started, proceed as follows:

1. Stop the pump.
2. Make sure pump is primed.
3. Suction lift may be too high. Check suction with a vacuum gage at pump suction. If suction lift is too high, the liquid may be vaporizing.
4. Check direction of rotation on a motor-driven unit.
5. Check pump and suction lines for air leaks.
6. Pump suction may be clogged.
7. Bypass relief valve may be lifting or held off seat by foreign matter.
8. Keep air vents open until pump discharges.

9470.256 PUMP NOT UP TO CAPACITY

If pump discharges but does not deliver sufficient capacity, one or more of the following principal causes may be responsible:

1. Air leaks in suction or through pump stuffing boxes.
2. Pump speed too low.
3. Suction lift too high.
4. Suction strainer, if fitted, may be too small or clogged.

5. End of suction pipe not submerged deep enough in liquid supply.
6. Pump damaged or badly worn or packing defective.
7. Relief valve improperly adjusted or held off seat. Bypass line to suction partly open.

9470.257 PUMP STARTS THEN CEASES TO DELIVER

If pump delivers liquid for a while and then loses its suction or fails to deliver, the following may be the cause:
1. Leaky suction lines.
2. Suction lift too high, causing liquid to vaporize.
3. Air or gas evolves into pockets in suction lines.

9470.258 PUMP TAKES TOO MUCH POWER

If pump takes too much power to operate, check the following:
1. Pump speed too high.
2. Liquid is heavier or more viscous than specified. Heating the liquid to reduce the viscosity should be undertaken if feasible. Reduce pump speed.
3. Suction or discharge lines are obstructed. Check pressures developed by pump. If no obstructions or partly closed valves are located, refer to the Naval Ships Systems Command giving complete data as to pressures, pump speed, viscosity, and temperature of liquid, and dimensioned sketch of system in use. Reduce pump speed to reduce load.
4. Mechanical defects such as bent shaft, binding of rotors, stuffing box packing too tight, misalinement on foundations or of piping connections resulting in distortion of casing and interferences within pump.

9470.259 PUMP IS NOISY

If pump is noisy, check the following points:
1. Air or gas in the liquid.
2. Air leaking into suction.
3. Liquid vaporizing in suction. Reduce suction lift.
4. Mechanical defects such as misalinement, rotor interference, high spots rubbing, or bent shaft.
5. Relief valve chattering.

9470.260 PUMP WEARS RAPIDLY

If pump wears excessively, check the following:
1. Liquid pumped contains sand or other abrasive and foreign matter.
2. Misalinement.
3. Pump operating at greater than rated pressure.
4. Operation on liquid of lower than specified viscosity.
5. Liquid is corrosive.
6. Pump has run dry or with insufficient liquid.

9470.261 VAPOR LOCK IN SUCTION LINE

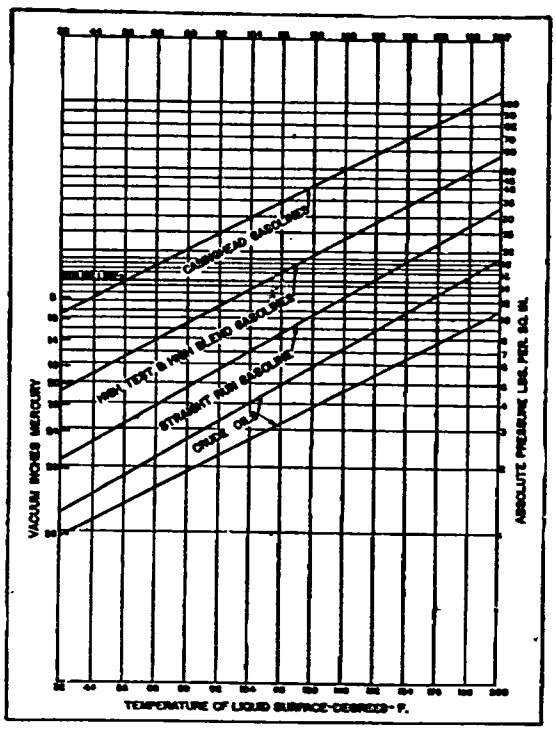
When a pump is in good condition and builds up a good vacuum on suction, but does not build up discharge pressure after all previously outlined possible troubles are corrected, the possibility of vapor lock in the suction piping should be investigated, particularly in long lengths of fuel oil suction piping, and especially after new oil cargo has been taken on or the oil has been heated up. First, shift suction to another tank or group of tanks nearer the pump. It will usually be found that the pump will readily build up pressure. If no nearer tank is available, build up an air pressure in the tank, taking particular care that pressure on the tank does not exceed the limits set by NAVSHIPS.

9470.262 VAPORIZATION OF OIL

1. The relationship between the temperature and vapor pressure of various petroleum products is shown in figure 9470-53. It can be seen that the lighter hydrocarbons such as gasoline have a much higher vapor pressure than the heavier fuel oils at the same temperature. If heavy pounding of the pump occurs when pumping fuel oil, it will usually be found that the vacuum on the suction side of the pump is excessively high or that the temperature of the oil is high for the type of oil being pumped causing partial vaporization in the suction line or pump casing.
2. Suction lift should not exceed a value such that the resulting absolute suction pressure is less than the sum of the vapor pressure at the pumping temperature plus 2 psi (approximately 4.1 inches of mercury). For example: A pump handling gasoline with a vapor pressure of 11 psi absolute should not have a suction lift exceeding 14.7−(11 + 2)=1.7 psi, or 3.5 inches of mercury suction lift.
3. Vapor lock is most apt to occur in pockets where the suction line overpasses other piping or ship's structure particularly in the proximity of the pump. It will be difficult for the operator to determine whether the maloperation resulting in noise and pounding is due to deaeration or vaporization but whatever is done to reduce the suction lift, such as reducing pump speed, will alleviate the condition.

9470.263 GAGES ON SUCTION

All fuel oil pumps should have compound gages installed at the pump suction manifolds in order to ascertain whether or not the suction lift vacuum is so great as to affect the pump capacity.



VAPOR PRESSURE CHART FOR DIFFERENT FUELS

Figure 9470-53

9470.264 PACKING GLANDS

Do not draw packing glands too tight. Packing should be kept adjusted so that there is a small amount of leakage of the liquid handled or of the sealing liquid. Watch stuffing boxes carefully on starting the pump. At the first sign of overheating, shut the pump down and let the stuffing boxes cool down before restarting. When stuffing box leakage is not permissible because of highly volatile and inflammable liquid pumped, the packing should be replaced frequently to avoid the packing scoring the shaft.

Part 3. Maintenance (where 3-M system is installed, conduct preventive maintenance in accordance with 3-M cards)

9470.281 REFERENCES

Instructions for the use of manufacturers' instruction books, as outlined in part 2 of this section, should be followed in the maintenance of rotary pumps. Instructions for care of shaft packing, lantern rings, shafts, lubricating and oiling system, bearings, thrust bearings, water cooling systems, couplings, and alinement as contained in section II, part 3 of this chapter, should be followed since they apply in general to rotary pumps as well as to centrifugal pumps.

9470.282 WEARING PLATES AND LINERS

The clearances between pump rotors and casing wearing plates and cylinder liners should be maintained as shown on manufacturers' plans. On low pressure, low suction lift pumps, such as lubricating oil, diesel oil supply, and tank drain pumps, the drop in pressure across the clearance spaces does not usually exceed 50 psi, and with this type of pump the clearance between parts may wear to as much as 0.005 to 0.010 inch without appreciable effect upon the capacity of the pump; hence, for this class of pump, wearing plate and liner renewal should not be undertaken until wear or increased clearance has exceeded these figures, or, in the case of fuel oil tank drain pumps, until the pump will not pull a vacuum of at least 16 inches mercury with suction valve closed. The engineering officer should decide in each case whether or not the amount of wear of clearance parts dictates renewal. For boiler fuel oil service pumps, however, the effect of increased clearance makes itself felt by reduction in the efficiency and in the maximum capacity of the pump. For this class of service the wearing parts should be renewed when the clearances exceed the designed clearances shown on manufacturers' plans by about 100 percent.

9470.283 THRUST BEARINGS

See comments concerning thrust bearings under section II, part 3 of this chapter. In pumps of the type shown in figures 9470-42 and 9470-43, the importance of the proper setting of the thrust bearings, which serve to hold the pumping elements centrally in the pump casing, cannot be overstressed.

9470.284 TIMING GEARS

In pumps of the type shown in figures 9470-39, 9470-40, 9470-42, 9470-43, and 9470-45, timing (or synchronizing) gears are fitted to the rotor shafts to maintain correct clearances between the two pumping rotors during operation. To accomplish this, the gears must be securely locked to the rotor shaft in their exact position to maintain a clearance between the rotor elements as the shafts make a complete revolution; therefore, no lost motion is permissible in way of keys or pins holding the rotors and gears in position on the shafts.

9470.285 MOTORS AND CONTROLLERS

The instructions for care of controllers and motors, chapter 9630, should be followed for motor-driven units.

9470.286 TURBINES

The instructions for care and operation of auxiliary turbines as contained in chapter 9500 should be followed where turbine-driven units are involved.

9470.287 PUMP PRESSURE GOVERNORS

Boiler fuel oil service, lubricating oil pumps, and some fuel oil booster pumps are fitted with adjustable constant pressure-type pump governors (chapter 9470 section IV) in addition to turbine governors mentioned in section II, part 3 of this chapter.

9470.288 TESTS AND INSPECTIONS (ROTARY PUMPS)

1. **Weekly.** Test operate idle pumps at normal conditions to inspect for mechanical seal leakage. Operate all pumps by steam or power. If power is not available, move by hand.
2. **Quarterly.**
   a. Lift all relief valves by water or oil as appropriate.
   b. Check thrust position of pump rotor.
   c. Sound and set up as necessary all foundation bolts and secure all foundation dowel pins.
3. **Annually.**
   a. Check bearing clearances by leads or crown thickness measurements.
   b. Check wear of internal parts by slowly throttling suction valve with pump running at normal speed. Do not close the suction valve completely as this may result in damage to the pump rotating elements and mechanical seal. A vacuum of at least 12 inches of mercury should be developed by fuel service pumps and 15 inches of mercury by the fuel booster and transfer pumps with NSFO and 5 inches of mercury should be developed with Distillate fuel by either fuel pump. Not less than 6 inches of mercury should be observed at the pump suction for lubricating oil service pumps. If pump will not pull the required vacuum, it should be opened, clearances measured, and necessary steps taken, either by renewal of parts, adjustment of rotors, or taking a cut off casing or cover as applicable, to restore designed clearances. In making this test be sure that pump is filled with oil before throttling the suction valve.
4. **Overhaul cycle.**
   a. Open pump and reduction gear casings; inspect and clean.
   b. Check clearances of all wearing plates and liners, casing throat bushing, rotors, casing liners, and bushings, and renew parts as necessary.
   c. Examine all pump rotors, shafts, bearings, timing gears, keys, and reduction gears, particulary worms and worm wheels.

NOTE: These tests and inspections are the minimum necessary to give adequate assurance of safe and reliable operation of equipment. Indications of low discharge pressure or other manifestations of improper operation should indicate more frequent or extensive tests and inspections. For tests and inspections of turbines, see chapter 9500, article 9500.47.

Part 4. Repairs

9470.301 ASSEMBLE DRAWINGS

When repairing or making an interior examination of a pump, it is essential that all drawings and available dimensional data be at hand. Frequently, such important dimensions as bridge gage readings, clearance between rotors and casing wearing plates and liners, shaft seal or gland adjustment became altered, resulting in poor operation. Such poor operation will naturally continue in spite of other major repairs unless the real cause of trouble is rectified.

9470.302 PACKING THE STUFFING BOX

The packing around the shafts of rotary pumps may be of stuffing box type, the labyrinth type, the rotary seal type, or a combination of types. As an additional precaution against entrance of air to the pump, seals filled with the liquid pumped are installed on the shaft. The shaft revolving at a high speed throws the sealing medium against the gland casing completing the seal. Failure to supply the packing of the stuffing box, or seals, with the sealing and lubricating liquid may cause the packing to become hot from excessive friction, seriously damaging the packing, and scoring the shaft or sleeve.

9470.303 RENEWAL OF WEARING PLATES AND LINERS

Whenever the pump casing is opened, the clearance between casing liners and plates, and various parts of the rotors and shaft, should be measured to see if excessive wear has taken place and if renewal is necessary. Should the bearings be worn excessively, it is reasonable to expect that the renewal of the liners is necessary. They should not be renewed, however, unless the pump bearings that may be down are first restored to their original readings. The required oil clearance for bearings is usually given on the manufacturers' plans. In the absence of such data, the table of tolerances and clearances (see chapter 9005) should be followed.

Part 5. Safety Precautions For Rotary Pumps

9470.311

1. See that relief valves, where fitted, are tested and that they function at the designated pressure.
2. Never attempt to jack a pump by hand while the steam valve to driving unit is open.
3. Do not tie down, or otherwise render inoperative, the overspeed trip, speedlimiting or speed-regulating governors.
4. Check setting of overspeed trip, if fitted, at least quarterly. See that overspeed trips are set to shut off all steam to the unit when rated speed is exceeded by 10 percent.
5. Check setting of speed-limiting and speed-regulating governors, where fitted, at least quarterly. See that they are set to limit the speed of the unit to rated speed under rated conditions, and that rated speed is not exceeded by more than five percent for any condition of loading.
6. Never operate a positive displacement rotary pump with discharge valve closed unless the discharge is protected by a properly set relief valve of size sufficient to prevent a dangerous rise in pressure.
7. When inspecting or overhauling vertical rotary pumps never rely on a flexible coupling to support the total weight of the pump rotors. Always support the pump rotors by means of wire rope slings, block and tackle, or chocks before working on pumps.

SECTION IV. PUMP PRESSURE-REGULATING GOVERNORS

Part 1. Description of Types and Services

9470.351 PRINCIPLE OF OPERATION

A pressure-regulating governor is essentially an automatic throttling valve installed in the steam supply line to a pump's driving unit. The governor is affected by the pump discharge pressure in such a manner that any variation in discharge pressure, or in pressure differential (article 9470.373) will actuate the governor causing it to regulate the pump speed by varying the flow of steam to the driving unit.

9470.352 GENERAL SERVICES

Pressure-regulating governors used in the service are applied almost exclusively to steam-driven pumps, both direct-acting reciprocating and turbine-driven rotary and centrifugal types. There are some pressure regulators fitted to motor-driven port and cruising fuel oil service pumps; however, regulators for this service are usually built as an integral part of the pump. It is suggested that, for the description and instructions relative to the care and operation of this latter type of pressure regulator, reference should be made to the manufacturers' instruction pamphlet for the pump.

9470.353 TYPES

There are normally two types of pressure-regulating governors provided, as follows:

1. A constant pressure (CP) type, which is designed to maintain a constant predetermined pump discharge pressure by control of the steam supply to the driving unit regardless of variation in steam pressure and load.
2. A differential pressure (DP) type, which is designed to maintain a constant predetermined difference in pressure between two selected points in the pump discharge line (or pump discharge line and boiler) by control of the steam supply to the pump driver, regardless of variation in steam pressure and pump load.

9470.354 APPLICATIONS, CP GOVERNORS

Constant pressure (CP) governors are applied to control the discharge pressure of fuel oil service pumps, fuel oil booster and transfer pumps, lubricating oil service pumps, fire and bilge pumps, main boiler feed pumps, and fire and flushing pumps.

9470. 355 APPLICATIONS, DP GOVERNORS

Differential pressure (DP) governors are applied mainly to the control of boiler feed pumps in order to maintain the desired pump output pressure.

9470.356 CLASSIFICATION

Pump pressure regulating governors are further classified as follows:

1. **Types**
   CP–Constant Pressure.
   CPAST–Constant pressure with automatic start.
   CPASP–Constant pressure with automatic stop.
   DP–Differential Pressure.
   DPAST–Differential pressure with automatic start.
   DPASP–Differential pressure with automatic stop.
2. **Classes**
   Class 1–Self-contained piston operated.
   Class 2–External pilot actuated.
   Class 3–Self-contained direct acting diaphragm.
3. **Series**
   Series 1500–Steam service 1500 psig max., 1000° F max.
   Series 600–Steam service 600 psig max., 850° F max.
   Series 600–Steam service 600 psig max., 750° F max.
   Series 300–Steam service 300 psig max., 750° F max.

9470.357 CP GOVERNOR DESCRIPTION

The design of constant pressure governor most used in service is the Leslie type (figure 9470-54).

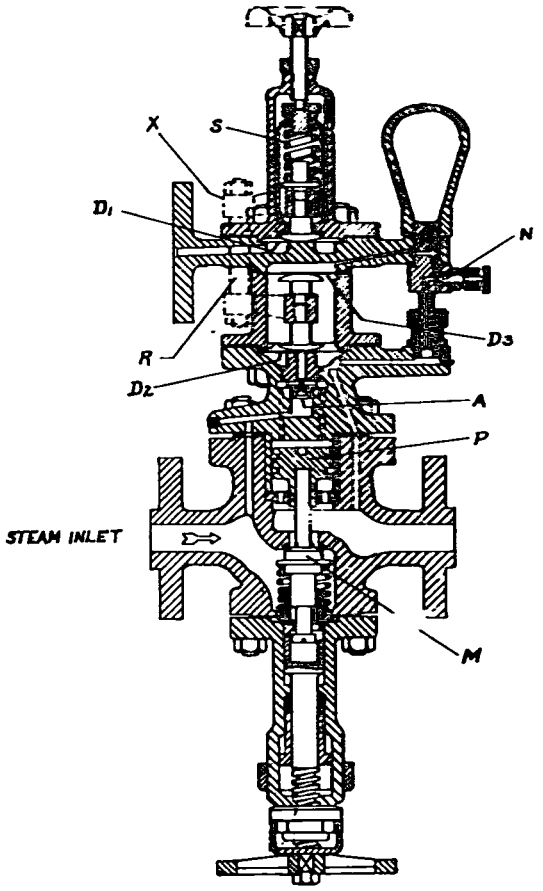


Figure 9470-54

9470.359 LESLIE CP GOVERNOR

1. The Leslie-type of constant pressure governor is illustrated in figure 9470-54. From an inspection of this figure, it will be noted that steam enters the left hand side of the valve, while the pump discharge pressure which is to be controlled is applied to the under side of the top diaphragm D_1 through a pressure-actuating line connected to the pump discharge. The force exerted on the under side of the top diaphragm is balanced by the pressure-adjusting spring S located at the top of the governor valve. Any movement of the top diaphragm is transmitted through the crossheads X and the connecting rods R to the lower diaphragm D_2, which in turn contacts and causes a corresponding movement of the pilot valve A. Operation is as follows: Steam from the inlet side of the governor is led through drilled passages to the under side of the pilot valve A. The pump discharge pressure, assumed to be lower than that for which the governor is set, acts on the under side of the top diaphragm D_1. As this pressure is not sufficient to overcome or balance the force of the pressure-adjusting spring S, the diaphragm and top crosshead X will move downward. This downward movement is transmitted through the connecting rods R and lower crosshead to the lower diaphragm D_2, which causes the pilot valve A to open, admitting steam from the inlet side to the top of the main valve operating piston P, which is forced down and opens the main valve M wider, thus admitting more steam to the pump and causing it to speed up and build up the discharge pressure. With pressure increase on the under side of the top diaphragm, the force acting under this diaphragm is increased sufficiently either to overcome or to balance the adjusting spring pressure, resulting in upward movement of the diaphragm, crossheads, and pilot valve, thus cutting off or reducing steam flow to top of the main valve operating piston, which causes the main valve to close partially and reduce pump discharge pressure to that for which the governor is set.

2. It will be noted that an intermediate diaphragm D_3 is fitted in the governor superstructure, the top of which is connected to the under side of the lower diaphragm through a passage fitted with a needle valve N. The under side of the lower diaphragm is in turn connected through passages to the steam outlet side of the governor valve. The purpose of this arrangement is to reduce hunting and erratic operation of the governor and also to improve regulation by causing the governor to respond more quickly to pump discharge pressure changes. The stabilizing needle valve and steam chamber between the upper and lower steam diaphragms are usually fitted on governors used in connection with turbine driven pumps. Turbine driven pumps operate at widely varying chest pressures and respond relatively slowly to steam pressure changes. Therefore, the governor must have a corresponding speed of response. The needle valve and steam chamber accomplish this by delaying the balancing of the diaphragms after a change in pump discharge pressure. This feature prevents over-regulation or hunting without reducing sensitivity.

3. Reciprocating pumps require a relatively constant steam chest pressure regardless of speed, and the governors have a simple, unrestricted connection between the upper and lower steam diaphragms maintaining constant balance.

4. A valve stem and handwheel are fitted below the main valve, the purpose of which is to open the governor

main steam valve, rendering the governor inoperative and allowing a full flow of steam through the governor. This device is intended to be used in case the governor fails to operate as intended. For normal operation the valve stem should be in its uppermost position to permit automatic operation of the governor. Where no pull-open device is provided, a bypass line is usually installed.

5. Some lubricating oil systems are supplied by a combination of turbine driven lubricating oil service pumps and a shaft driven pump. To permit the independent turbine-driven pumps to remain warm and in a condition to immediately supply oil to the system if required, the turbine driven pump pressure governors have a one-eighth inch diameter hole drilled in the valve disc, permitting the turbine driven pumps to idle at about 20 revolutions per minute.

9470.360 AUTOMATIC SHUTDOWN DEVICE

The Leslie Company has developed an auxiliary pilot valve for use in conjunction with constant pressure governors on boiler feed systems which have separate main feed and booster pumps. The assembly of the auxiliary pilot valve and the governor which has a special top cap has been termed an automatic shutdown device. Figure 9470-55 shows the arrangement of this device. Figure 9470-56 shows the auxiliary pilot valve. The function of the auxiliary pilot valve is to control the flow of steam to the top side of the governor piston which operates the main valve. The special top cap takes actuating steam from the auxiliary pilot valve instead of through the main body as in the conventional top cap. Otherwise the governor functions in normal fashion. It will be noted that the auxiliary pilot valve has an actuating line to the booster pump discharge. The purpose of this device is to protect the main feed pump against damage in event of loss of booster pump pressure. The automatic shutdown devices have demonstrated on service tests that they will shut the main feed pump down in event of loss of or inadequate booster pump pressure. The latest design of the automatic shutdown device combines the functions of the two units into a single governing valve with automatic trip feature. Figure 9470-57 is typical of such an installation for a system using a series 1500 governor.

9470.361 DP GOVERNORS

There are relatively few differential pressure governors in service and the discussion will be limited to a typical example of this type of governor.

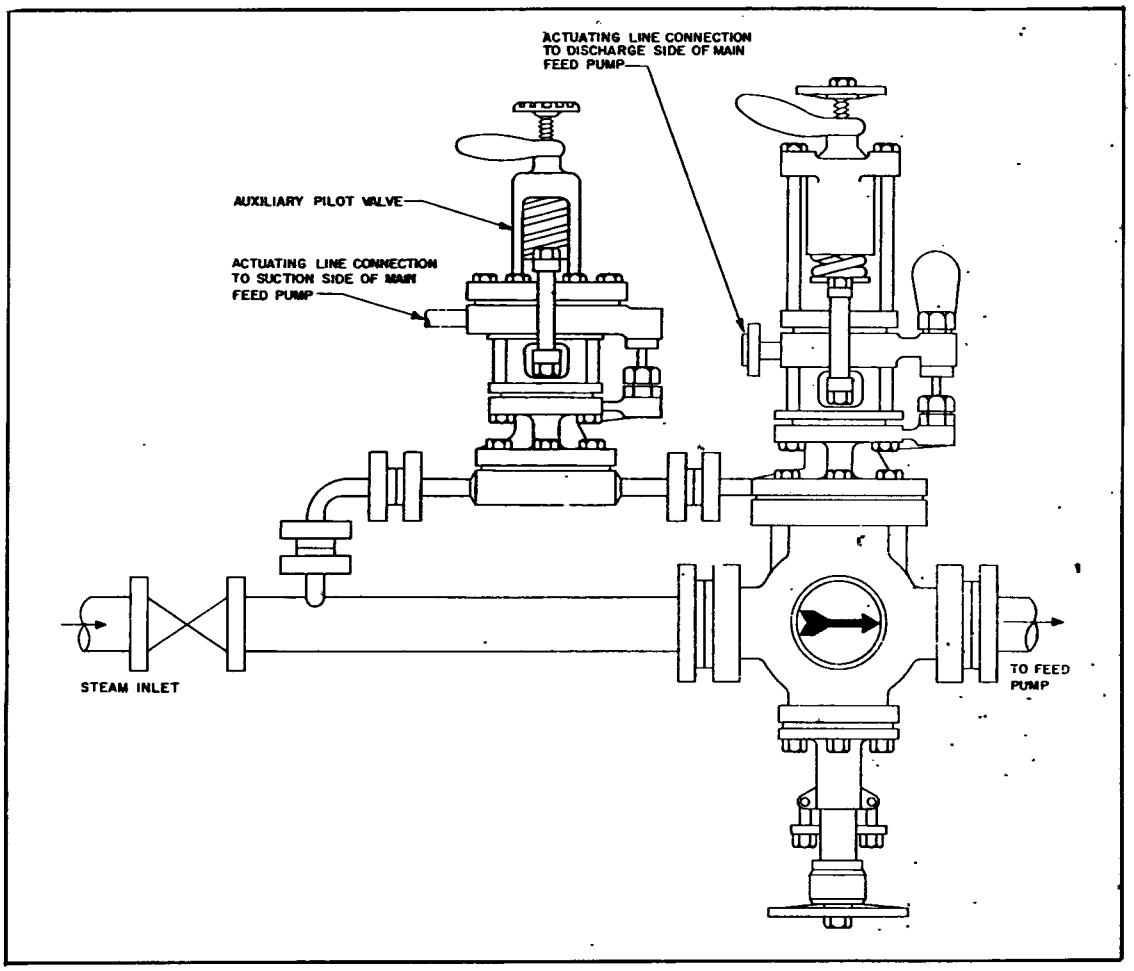

ACTUATING LINE CONNECTION TO DISCHARGE SIDE OF MAIN FEED PUMP
AUXILIARY PILOT VALVE
ACTUATING LINE CONNECTION TO SUCTION SIDE OF MAIN FEED PUMP
STEAM INLET
TO FEED PUMP

Figure 9470-55

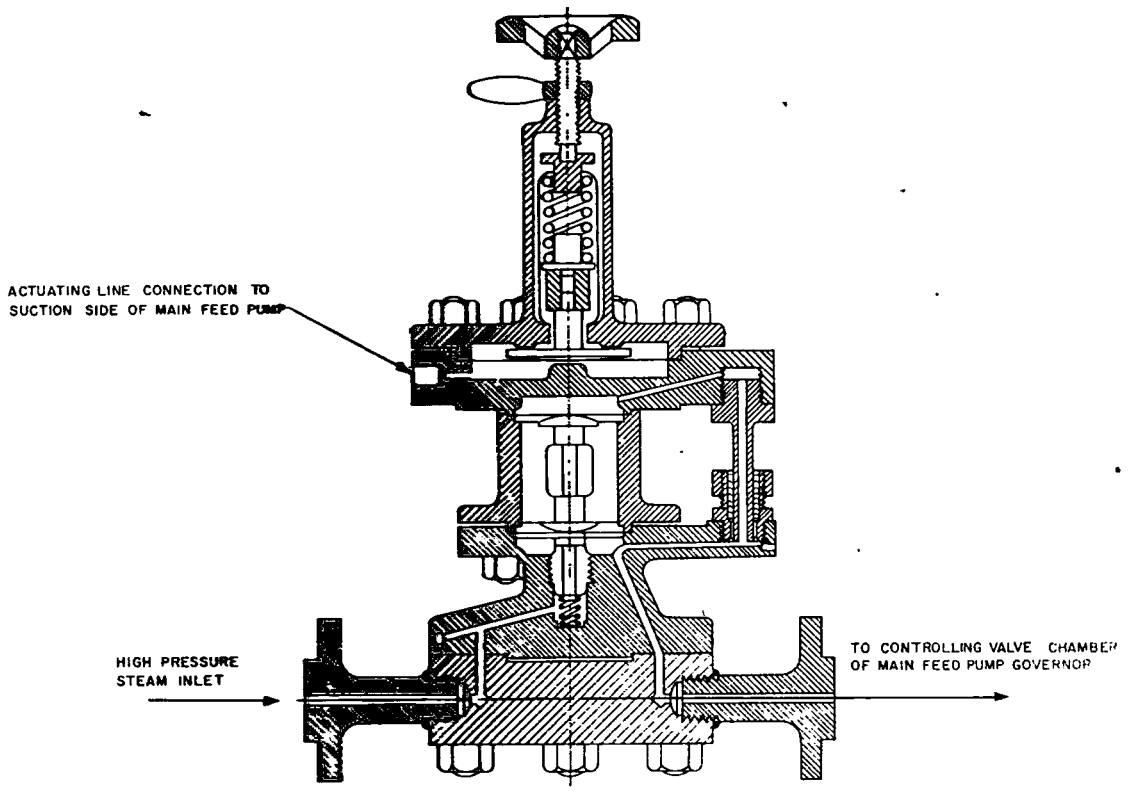


Figure 9470-56

1. The Leslie differential pressure governor is similar in design to the constant pressure type except that an additional diaphragm is fitted at the top of the superstructure to sense the difference in pressure between two selected points in the pump discharge line or pump discharge line and boiler pressure.

2. The lower pressure acting downward on the top diaphragm moves the external yoke downward and opens the controlling valve. This valve admits high pressure steam from the inlet body port to the top of the piston, opening the main valve and admitting steam pressure to the turbine. The pump discharge pressure acts upward under its diaphragm and balances the lower pressure. Compressing the adjusting spring by the handwheel adds to the downward force of the lower pressure, thereby increasing the pump discharge pressure necessary to balance it. When the pump discharge pressure balances the lower pressure plus the spring force, the controlling valve throttles so that the necessary volume of steam is delivered to the turbine to maintain this constant differential pressure. Ordinary fluctuations in the inlet pressure do not change the differential pressure setting; only a change in the differential pressure affects governor action.

9470.362 LESLIE GOVERNOR SYMBOL DESIGNATIONS

All Leslie governors are classified by certain symbol designations and since these governors are used most extensively in service, the following information will assist in their ready identification:

P = Constant pressure under 400 pounds.
T = Turbine
L = Low pressure--150 pounds (maximum) controlled pressure.
N = Navy (United States).
S = Steel.
H = High pressure--150 pounds (minimum) controlled pressure.
X = Differential pressure.
R = Reciprocating.
C = Constant pressure over 400 pounds.
Z = Low pressure suction trip.

Number after symbol means modification. No "S" in class indicates bronze body.

Part 2. Operation

9470.371 LESLIE PRESSURE REGULATING GOVERNORS

1. Instructions for cutting in governor:

a. Set the manual pull-open device on the main valve for automatic operation.

b. Open the shutoff valves in the steam and liquid actuating lines to the pressure governors.

c. Open steam throttle valve gradually until further opening of throttle valve causes no further increase in speed or discharge pressure of the unit. (See instructions for starting various pumps as contained in sections I, II, and III of this chapter.)

d. Operate under this condition for a few minutes to make certain that the governor is operating satisfactorily, then open the throttle valve wide.

e. For constant pressure governors, adjust the handwheel so that the desired pump discharge pressure is obtained. Screw the handwheel downward to increase and upward to decrease pressure. Lock with lock nut.

NOTE: If the governor is shut down for maintenance, the inlet root valve is closed and the turbine exhaust valve secured. Any steam in the turbine casing is bled out through the turbine casing drain valve.

f. For excess pressure governors, adjust the handwheel so that the pump discharge pressure is the desired amount above the boiler pressure. Screw the handwheel downward to increase and upward to decrease the differential between pump discharge pressure and boiler pressure. Lock with lock nut.

g. If the governor surges or hunts, throttle the needle valve (N) (if fitted) in the equalizing passage connecting the under side of the bottom diaphragm with the upper side of the intermediate diaphragm (Figure 9470-54) until discharge pressure becomes steady. Never shut the needle valve tight. When pumping minimum flow, adjust the needle valve approximately 1/4 turn open so that there is no "hunting" and the governor action is smooth and even.

NOTE: In some makes of governors the throttling effect required to steady the pressure is inherent in the governor design and no needle valve will be found on the governor.

h. If the governor fails to control the pump discharge pressure properly, it should be made inoperative in the open position by the manual pull-open device on the button of the governor. Then control the pump discharge pressure by the hand throttle valve.

NOTE: Some makes of governors and those of early design have no manual pull-open device; in such cases the governor can be forced wide open by shutting off the liquid actuating line, and increasing the adjusting spring pressure by means of the hand wheel.

2. Instructions for cutting out a governor:

a. If the pump is to be kept in operation, choke the throttle valve until the pump starts to slow down; close the steam throttle valve if the pump is to be secured.

b. Close the steam actuating line valve.

c. Close the liquid actuating line valve.

d. Throw the governor valve wide open by means of the pull-open device, if fitted.

e. Open drain valve (where fitted) if pump is to be secured.

9470.372 LESLIE PRESSURE REGULATING GOVERNOR WITH LOW PRESSURE SUCTION TRIP–OPERATION INSTRUCTION

Instructions for cutting in feed pump governor with integral low pressure suction trip:

1. Follow general preliminary pump start-up procedures as outlined by pump manufacturer for the particular pump, making sure that the pump recirculating and discharge lines are open.

2. Before starting pump pressure governor, have all compression off the adjusting spring by turning upper handwheel counterclockwise and lower handwheel all the way clockwise. Open the stop valves in steam supply and actuating lines fully. Allow a short time for condensate reservoir of bottom cap to fill with condensate.

3. Supply suction pressure to actuator diaphragm of trip device. Engage trip mechanism by lifting trip gag pin and pushing the handle inward until pin enters hole for automatic operation. Turn upper handwheel slowly clockwise until desired pressure is obtained. Tighten half wing locknut.

4. Adjust trip device for the correct trip-out pressure by screwing the adjusting nut upward to compress actuator adjusting spring until the correct trip-out pressure is reached. Test by varying the suction pressure. Recommended trip-out pressure is 40 psig (unless otherwise specified).

5. Speed limiting governor of turbine must be set at a correct value above maximum operating so as not to interfere with pump governor as specified in the pump manufacturer's instruction book.

6. Handwheel at lower end of pump governor is for emergency operation only. By screwing the stem out counterclockwise, main valve is pulled off it's seat when indicator plate reads "open." Normally, the stem must be all the way in so that indicator plate will read "auto" when pump governor works automatically.

7. In the event of loss in suction pressure on the actuator diaphragm, trip mechanism will operate to close pump governor.

8. When suction has been returned to normal, the trip mechanism must be manually returned to automatic operation position by pressing trip reset handle down.

9470.373 PERFORMANCE REQUIREMENTS

1. Centrifugal boiler feed pumps.

a. For centrifugal boiler feed pumps operating under steady steam pressure and with the governor set for the rated discharge pressure at rated capacity, the pump discharge pressure shall rise from rated capacity of the pump to shut-off by an amount not less than 2.5 percent nor more than 4.0 percent. This rise in pressure shall be continuous; that is, at no point of the curve from rated capacity to shut-off shall there be a drop in the characteristic. (This requirement is based on pumps having constantly rising head-capacity curves with shut-off, at constant rated speed, not less than 15 percent greater than rated discharge pressure.) The pressure-capacity characteristic curves of the pump, with its governor in control, when determined from rated capacity to shut-off and from shut-off to rated capacity, shall not vary at any given capacity by an amount in excess of 0.67 percent of the rated discharge pressure for pumps rated at 750 psig or less, and 1.0 percent for pumps rated at more than 750 psig.

b. Under steady operating conditions the constant pressure for which the governor is set shall not vary more than plus or minus 0.67 percent of the governor setting. Maximum momentary variation in pump discharge pressure due to sudden major changes in capacity shall not exceed plus or minus 10 percent of the rated pump discharge pressure, and such fluctuations shall settle down to normal within 10 seconds. During a gradual reduction in steam

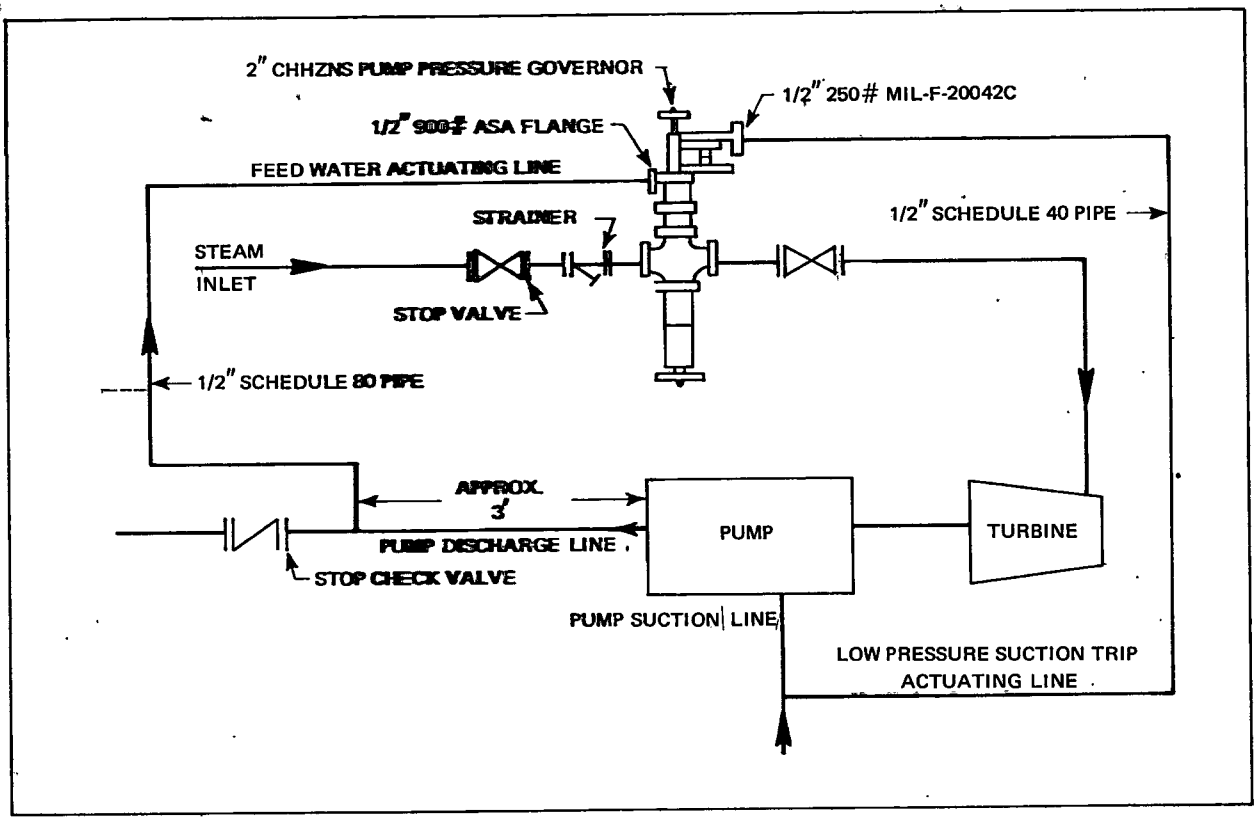


Figure 9470-57. Schematic Piping Diagram

pressure of 20 percent, and vice versa, the pump discharge shall not vary more than plus or minus 1.5 percent with turbine overload nozzle open.

c. For requirements as to paralleling and load distribution for boiler feed pumps equipped with constant pressure governors, see section II.

2. Fuel oil service pumps.

a. Under steady operating conditions the pump discharge pressure shall not vary more than plus or minus two percent.

b. Under steady normal operating conditions with any setting of pump discharge pressure within the specified range of adjustment, the pump discharge pressure shall rise not less than 2.8 percent nor more than 5.7 percent during gradual changes from full load to 25 percent load and vice versa, and under these conditions the governor shall be stable from full load to minimum load and vice versa. During a gradual reduction in steam pressure of 20 percent and vice versa, the pump discharge pressure shall not vary more than plus or minus 2.8 percent of the rated discharge pressure.

c. Maximum momentary variation in pump discharge pressure due to sudden major changes in capacity with steady steaming conditions shall not exceed plus or minus 10 percent of the governor pressure setting and such fluctuations shall settle down to normal within 10 seconds.

3. Other pumps.

a. Under steady operating conditions the pump discharge pressure shall not vary more than 1 p.s.i. total for LO service, and plus or minus 2 percent, or 5 p.s.i., whichever is greater, for FS service.

b. Under normal operating steam conditions, with any setting of the governor, the pump discharge pressure shall be maintained within a tolerance of plus or minus five p.s.i. for series 300 and 600 governors, and 7.5 p.s.i. for series 1500 governors during gradual changes from 25 percent load to full load and vice versa, also during a gradual reduction in steam pressure of 20 percent and vice versa. The same pump discharge pressure tolerances shall be maintained.

c. Maximum momentary variation due to sudden major changes in capacity with steady steam conditions shall not exceed plus or minus 10 percent of the pump discharge pressure for pressures above 100 psig or above or plus or minus 10 p.s.i. for lower pressure; and such fluctuations shall settle down to normal within 10 seconds.

Part 3. Installation, Care and Repair

9470.381 USE OF MANUFACTURERS' PAMPHLET

The instructions for installation, operation, care, and repair of pressure governors contained in this chapter are general for all makes and types of governors. The manufacturers of the various types of governors issue pamphlets which contain specific instructions for each type of governor. Engineer officers should see that these pamphlets are on board and should use them in conjunction with these instructions. (See comments under section II of this chapter.)

9470.382 LIQUID AND STEAM GOVERNOR ACTUATING PIPES

The actuating pipe lines for various services should in all cases be connected as follows:

1. For boiler feed service:

a. (Constant pressure governor only.) The water actuating line should be taken from the discharge of the pump between the pump discharge flange and discharge check valve for centrifugal pumps, and from the discharge of the pump between the discharge flange and the discharge stop valve for reciprocating pumps.

b. (Differential pressure governor only.) The actuating lines sense the difference in pressure between two selected points in the pump discharge line or pump discharge line and boiler pressure. Refer to 9470.361.

2. For boiler fuel oil service (constant pressure governor only), the oil actuating line should be taken from the discharge of the pump beyond the check valve between the pump and discharge stop valve.

3. For fuel oil booster service (constant pressure governor only), the oil actuating line should be taken from the discharge of the pump between the pump discharge flange and the discharge stop valve.

4. For lubricating oil service (constant pressure governor only), the oil actuating line should be taken from the lubricating oil main beyond the lubricating oil discharge strainer and the lubricating oil cooler, except when otherwise shown on approved plans.

5. For fire pump service (constant pressure governor only), the water actuating line should be taken from the discharge of the pump between the pump discharge flange and its discharge check valve.

9470.383 CONTROL VALVES AND SEATS

1. The control or pilot valve is continuously throttling or wiredrawing the steam passing through it, and hence the valve and valve seat are subjected to considerable erosion; therefore, they should be inspected periodically. New parts when needed should be of hardened CRES steel as detailed in manufacturer's drawings.

2. Incident to the Leslie governors some difficulties have been experienced in reassembling the control valve assembly within the top cap and maintaining a tight joint between the control valve seat on the top cap casting. The following instructions are given to remedy this difficulty: If the thread fit of the valve seat in the top cap does not seem to be tight enough to prevent the valve seat from backing out, it should be staked or otherwise locked in place in the cap. The following important points should be noted in cleaning and reassembling the top cap assembly (figure 9470-58):

a. In order to obtain continually satisfactory service, the joint "X" between the controlling valve seat and top cap must be maintained perfectly steam tight at all times.

b. Clean all parts, including threads with solvent.

c. Polish joint faces at "X" with crocus cloth for metal-to-metal joint.

d. When installing controlling valve seat, tighten firmly with wrench furnished for this purpose, and, when all the way down, tap wrench to ensure dead tight joint.

e. Correct clearance between top of controlling valve stem and diaphragm seat is 0.001 inch to 0.002 inch.

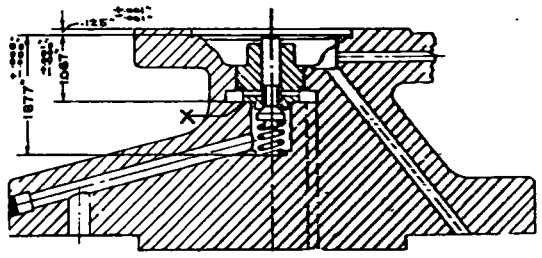


Figure 9470-58

f. If necessary to remachine top cap, tolerance given must be adhered to. Joint face "X" must be square with axis of thread and have straight smooth finish without any tool marks.

9470.384 CONTROL VALVE SPRING

Control valve springs should be examined occasionally to see that they are in good condition. In some of the earlier governor designs, the space around the spring was not properly drained of condensate and frequent breakage of the spring resulted.

9470.385 MAIN VALVES AND SEATS

Governors are inherently throttling devices; hence, main steam valve will be subject to erosion due to wire drawing of steam. Where erosion is excessive, the valves and seats should be replaced with materials as prescribed above for control valves and seats. However, new main bodies (steel) are furnished with stellited seats welded integral with the body. In this instance only the main valve is renewable.

9470.386 PISTON AND CYLINDER LINERS

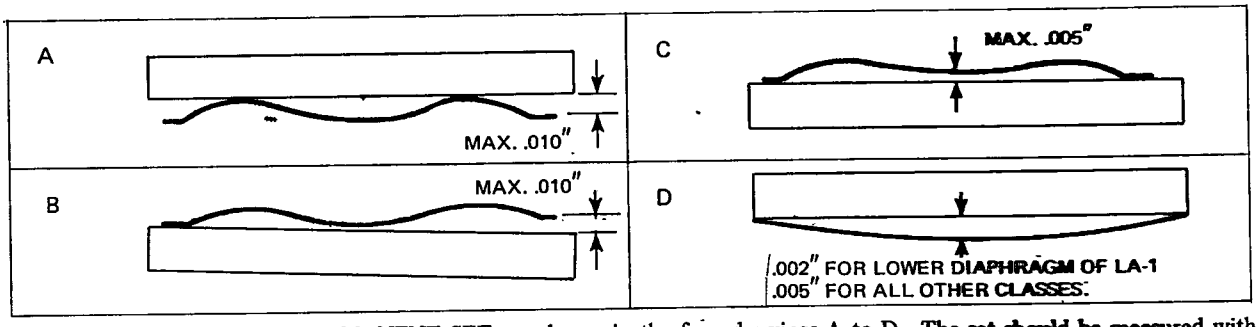
The main valve operating pistons and cylinder liners frequently cause faulty governor action due to accumulation of foreign matter which causes sticking or to excessively worn cylinder liners. The latter is frequently the result of wearing of grooves in the liner due to the movement of the piston through a short travel, as when the pump is operated at reduced capacity. Then, when the governor attempts to open the main valve wide for full pump capacity, the travel of the valve is limited to the width of the grooves, because the piston rings cannot get past the ridges. Thus the governor fails to open wide. This is frequently caused by the cylinder liner being softer than the piston and piston rings. Where this condition is found or where replacement of parts is necessary, the cylinder liner should be made of corrosion-resisting steel, grade 5, hardened to 475 Brinell, the piston of CRES 400 series and the piston rings of high test grey iron. Liners of CRES steel should be substituted for bronze cylinder liners which were installed as a war time substitute for the proper material in most steel governors.

9470.387 DIAPHRAGMS

1. Whenever governors are disassembled, all diaphragms should be carefully inspected. If indications of incipient failure are found, the affected part should be renewed. However, relative to the routine overhaul of Leslie pump governors, many serviceable diaphragms are being replaced, due to the fact that the removed diaphragms are distorted as a result of normal operating pressure.

2. Distortion of diaphragms, within certain limits, is normal, and such distortion has no adverse effect on the operation of the governor. Figure 9470-59 shows the permissible distortion for various types of diaphragms. Unless distortion beyond the limits indicated is found, replacement of diaphragms is not necessary.

3. The importance of replacing diaphragms with ones of the proper thickness and of specified material is stressed. Unless manufacturer-furnished spares are used, reference should be made to finished plans for guidance as to proper dimensions and materials. For further information there is listed in figure 9470-59 diaphragm dimensions and materials used on various types of Leslie regulators.



PERMISSIBLE LIMITS OF PERMANENT SET are shown in the four drawings A to D. The set should be measured with straight edge and feeler gages.

TABLE FOR DIAPHRAGMS USED IN LESLIE REGULATORS

LESLIE CLASS	REFERENCE NUMBER	THICKNESS Inch	DIAMETER Inches	MATERIAL
LT, LT-1, LT-3, LTP-3, J, J-1, JA, L, L-1, L-2, L-3, LE-3, LN, (E, EK, EKX, F, FK up to 1-1/2") L-1A, LE-3A, L-3A, EK-42	4552	0.025	2-33/64	CRES 302
EK, EK-28, FK, FK-28 (over 2")	Disc.	.032	3-3/32	CRES 302
EK-28RI, EKL, FK-28RI	Disc.	.020	4-5/16	CRES 302
L-RI, L-1, RI, LL, LL-3, LT-RI	4995	.020	3-3/32	CRES 302
LTC, JTC, JL, JAL	12007	.0125	2-33/64	CRES 302
LTCO JTCO	9120	.020	2-33/64	CRES 302
LT, LT-1, LT-RI, LT-3, LTP-3, LTL-3 (upper diaphragm)	9945	.025	5	
PTH, PRH, PTL, PRL, LS, LL-S, LB, LX, LB-1, PR, CTH, CRH, XTH, XRH	4552	.025	2-33/64	CRES 302
PTL, PRL, PTL(N)S, PRL(N)S (upper diaphragm)	9814	.025	4-5/16	Monel
LAS, CTH(N)S, CRH(N)S, PTH(N)S, LAS-2, PTL(N)S, PRL(N)S, XTH(N)S XRH(N)S, LA, LA-1, LAS-3, LNS-2	10425	.032	2-33/64	CRES 302
LTCS	12007	.0125	2-33/64	CRES 302
LTCSO	9120	.020	2-33/64	
EKX-42	10425	.032	2-33/64	
PLNS, PLNS-3, PLNS-13 (diaphragm upper)	14090	.007	7-1/4	Phos. Brz.
PTLNS-13 (upper diaphragm) (two leaf diaphragm each .006)	14270	.012	4-5/16	Phos. Brz.
LTCL	4787	.0125	3-3/32	CRES 302
LTCLS	Disc. 4993	.018	3-3/32	CRES 302
1/2" to 1-1/2" DF, DFS	13246	1/16	9-3/4	Rubber
2-1/2" to 5" D-1, DS-1 & 2" to 6" DF, DFS	13247	1/16	14-3/4	Rubber
6" D-1	16086	1/16	14-3/4	Rubber
PD & PR diaphragm for nozzle	12368	.025	1-47/64	Neoprene
PD & PR diaphragm for nozzle disc	12369	.007	3.091	Phos Brz.
Diaphragm upper, control Pilot 1/2 to 6 lbs.	13004	1/16	8-1/8	Neoprene
Diaphragm upper, control Pilot 2 to 20 lbs.	12476	3/64	4-31/32	Neoprene
Diaphragm upper, control Pilot 15 75, 50 to 125 lbs.	13257	3/64	4-15/32	Neoprene
Diaphragm upper, control Pilot 100 to 200, 175 to 300, 275 to 450 lbs.	13226	1/16	3-7/32	Neoprene
Diaphragm for thermostats D, D1, R, R1	10895	.008	1.872	Phos Brz.

Figure 9470-59

CHAPTER 9470 – PUMPS

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